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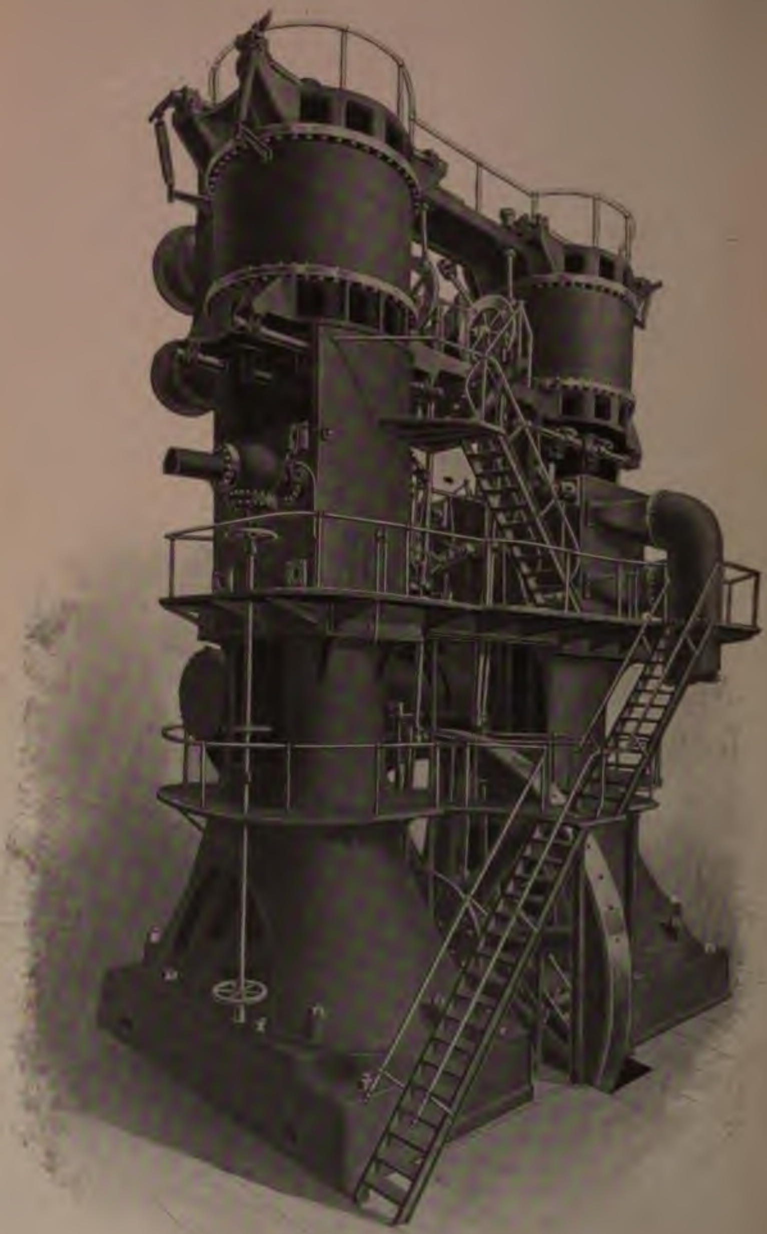

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Introductory Note

THE Cyclopedia of Engineering is compiled from the most practical and comprehensive instruction papers of the American School of Correspondence. It is intended to furnish instruction to those who cannot take a correspondence course, in the same manner as the American School of Correspondence affords instruction to those who cannot attend a resident engineering college.

¶ The instruction papers forming the Cyclopedia have been prepared especially for home study by acknowledged authorities, and represent the most careful study of actual shop needs and conditions. Although primarily intended for correspondence study, they are in use as text books by Columbia University, Lehigh University, Iowa State College, the University of Maine, the U. S. Government in its School of Submarine Defense, the Westinghouse Companies in their Shop School, and for reference in the leading colleges, shops and public libraries.

¶ Years of experience in the shop, laboratory, and class-room have been required in the preparation of the various sections of the Cyclopedia. Each section has been tested by actual use for its practical value to the man who desires to know the latest and best practice in the shop or engine room.

¶ Numerous examples for practice are inserted at intervals; these, with the test questions, help the reader to fix in mind the essential points, thus combining the advantages of a text book with a reference work.

¶ Grateful acknowledgment is due to the corps of writers and collaborators, who have prepared the many sections of this work. The hearty co-operation of these men—engineers of wide practical experience and acknowledged ability — has alone made these volumes possible.

¶ The Cyclopedia has been compiled with the idea of making it a work thoroughly technical yet easily comprehended by the man who has but little time in which to acquaint himself with the fundamental branches of practical engineering. If, therefore, it should benefit any of the large number of workers who need, yet lack, technical training, the editors will feel that its mission has been accomplished.

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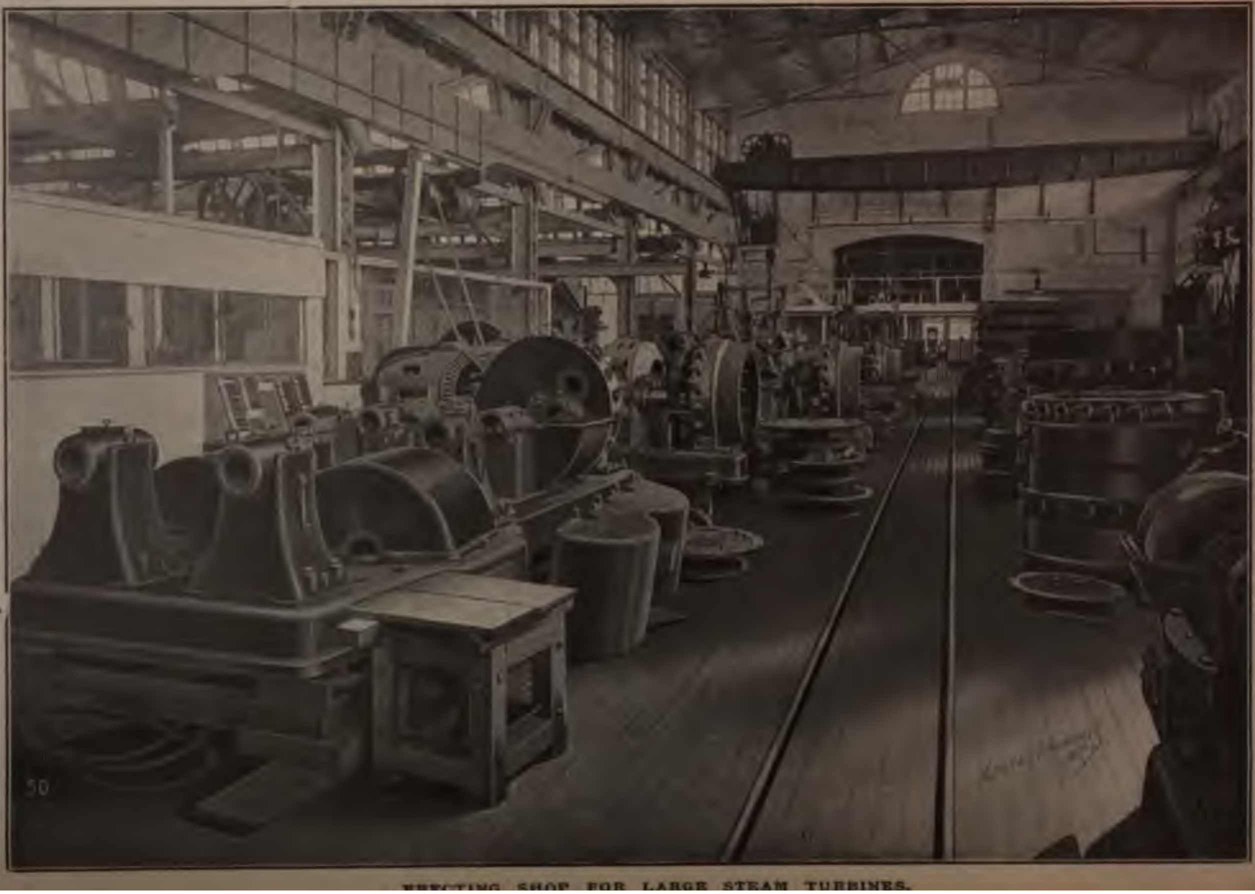


ERECTING SHOP FOR LARGE STEAM TURBINES.

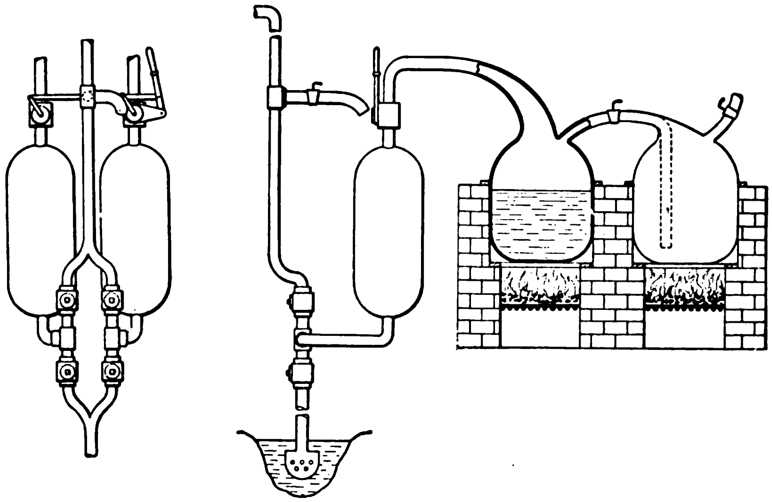
THE STEAM ENGINE.

There are various kinds of engines from which mechanical work is obtained by the expenditure of heat. In the gas engine a mixture of gas and air is burned in the cylinder, the heat thus generated being converted into work by the expansion of the products of combustion. The action in oil and hot-air engines is very similar. The most important of all heat engines, however, is the steam engine, in which the heat in steam is transformed into work. It will be useful to review briefly some of the stages through which it has passed in its development.

The first steam engines of which we have any knowledge were described by Hero of Alexandria, in a book written two centuries before Christ. Some of them were very ingenious, but the best were little more than toys. From the time of Hero until the seventeenth century there was very little progress. At this time there began to be great need of steam pumps to remove water from the coal mines. In 1615, Salomon de Caus devised the following arrangement. A vessel, having a pipe leading from the bottom, was filled with water and then closed. Heat applied to the vessel caused steam to be formed, which forced the water through the pipe.

A little later an engine was constructed in the form of a steam turbine; but it was unsuccessful, and the attention of inventors was again turned to pumps.

Finally Thomas Savery completed, in 1693, the first *commercially successful* steam engine. It was very wasteful of steam as compared with our engines of today, but as being the first engine to accomplish its task it was a grand success. Savery's engine (Fig. 1) consisted of two oval vessels placed side by side and in communication with a boiler. The lower parts were connected by tubes fitted with suitable valves. Steam from the boiler was admitted to one of the vessels and the air driven out. The steam was then condensed and a vacuum formed by letting water play over the surface of the vessel. When the valve opened, this vacuum drew water from below until the vessel was full. The valve was then closed and steam again admitted, so that on opening the second valve the water was forced out through the delivery pipe. The two vessels worked alternately. When one was filling with water, the other was open to the boiler and was being emptied. Of the two boilers, one supplied steam to the oval vessels and the other was used for feeding water to the first boiler. The second boiler was filled while cold and a fire lighted under it. It then acted like the vessel used by Salomon de Caus and forced a supply of feed water into the main boiler.



END VIEW. Fig. 1. SIDE VIEW.

A modification of Savery's engine, the pulsometer (Fig. 2), is still quite common. It is used in places where an ordinary pump could not be used and where extreme simplicity is of especial advantage. Its valves work automatically and it requires very little attention.

A serious difficulty with Savery's engine resulted from the fact that the height to which water could be raised was limited by the pressure which the vessels could bear. Where the mine was very deep it was necessary to use several engines, each one raising the water a part of the whole distance. The consumption of coal in proportion to the work done was about twenty times as great as that of a good modern steam engine. This was largely, though not entirely, due to the immense amount of steam which was wasted by condensation when it came in contact with the water in the oval vessels.

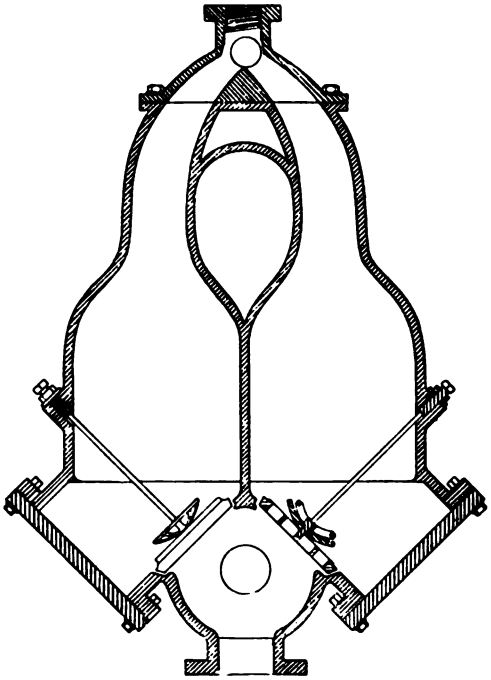


Fig. 2.

The next great step in the development of the steam engine was taken by Newcomen, who in 1705 succeeded in preventing contact between the steam and the water to be pumped, thus diminishing the amount of steam uselessly condensed. He introduced the first successful engine which used a piston working in a cylinder.

In Newcomen's engine, shown in Fig. 3, there was a horizontal lever pivoted at the center and carrying at one end a long heavy rod which connected with a pump in the mine below. A piston was hung from the other end of the lever, and worked up and down in a vertical cylinder, which was open at the top. Steam acted only on the lower side of the piston. Steam at atmospheric pressure was admitted from the boiler to the cylinder, and as the pressure was the same both above and below the piston, the falling of the heavy pump rod raised the piston. A jet of water was now passed into the cylinder to condense the steam and form a vacuum. This left the piston with atmospheric pressure above and very slight pressure below, so it was forced down and the pump rod again raised. Steam could again be admitted to the cylinder, the pump rod would fall, and so on indefinitely.

In the days of Newcomen it was very difficult to obtain good workmanship. For this reason it was often necessary to make the cylinders of wood, and even then there might be a space of one-eighth of an inch between the wall of the cylinder and the piston. In order to prevent steam from blowing through this passage, or air from leaking in when the steam was condensed, it was customary to keep a jet of water playing on the top of the piston.

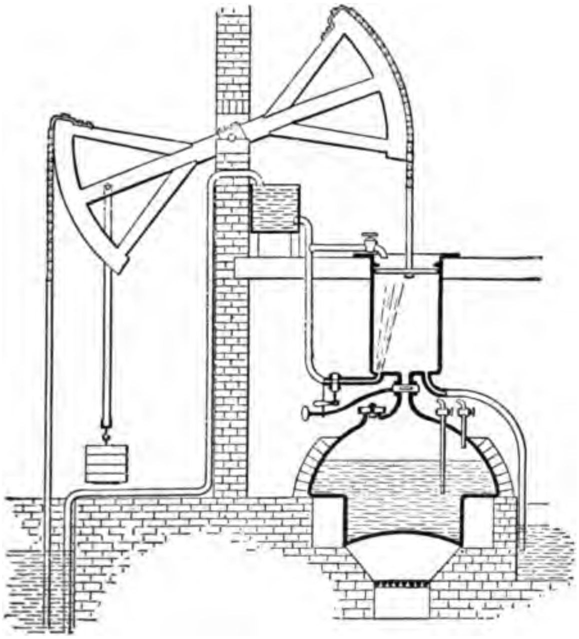


Fig. 3.

One great trouble with all these engines was that they required some one to open and close the cocks. Boys were generally employed to do this work. In order to get time to play, one of them rigged a catch at the end of a cord which was attached to the beam overhead. This did the work for him. Making the valves automatic in this way made it possible to dispense with the services of the boy and at the same time greatly increase the speed of the engine. This engine was improved slightly from time to time by different inventors and was very extensively used until

Watt's time. Some of them are in existence today. While this engine was a success and a great improvement over its predecessors, it was still very large, wasteful and heavy, in comparison with the work done. When the cylinders were made of iron they were simply cast and not bored, thus leaving a rough, stony coating over the iron, called the skin.

In the year 1763, a small model of a Newcomen engine was taken to the shop of an instrument maker in Glasgow, Scotland, to be repaired. This instrument maker, whose name was James Watt, had been studying steam engines for some time and he became very much interested in this model. He was a man of great genius, and before he died his inventions had made the steam engine so perfect a machine that there has been but one really great improvement in it since his time; namely, compound expansion. All other improvements have been merely following in the line of his suggestions and constructing what he could not for lack of good tools.

He found that to obtain the best results it was necessary, "*First, that the temperature of the cylinder should always be the same as that of the steam which entered it; and, secondly, that when the steam was condensed it should be cooled to as low a temperature as possible.*" All improvements in steam-engine efficiency have been in the direction of a more complete realization of these two conditions.

In order to keep the cylinder nearly as hot as the entering steam, Watt no longer injected water into the cylinder to condense the steam, but used a separate vessel or condenser. He made his piston tight by using greater care in construction, so that it was not necessary to have a water seal at the top. He then covered the top of the cylinder to prevent air from cooling the piston. When this was done he could use steam above as well as below the piston; this made the engine double acting.

Also, in the effort to keep the cylinder as hot as the entering steam, he enclosed the cylinder in a larger one and filled the space between with steam. This was not often done, however, and only of late years has the steam jacket been of much advantage. He also used steam expansively, that is, the admission of steam was stopped when the piston had made a part of the stroke; the rest of the stroke was completed by the expansion of the steam already admitted. This plan is now used in all engines that are built for economy.

Other inventions made by Watt on his steam engine were: a parallel motion, that is, an arrangement of links connecting the end of the piston rod with the beam of the engine in such a way as to guide the rod almost exactly in a straight line; the throttle valve, for regulating the rate of admission of steam and the centrifugal governor, which controlled the speed by acting on the throttle valve.

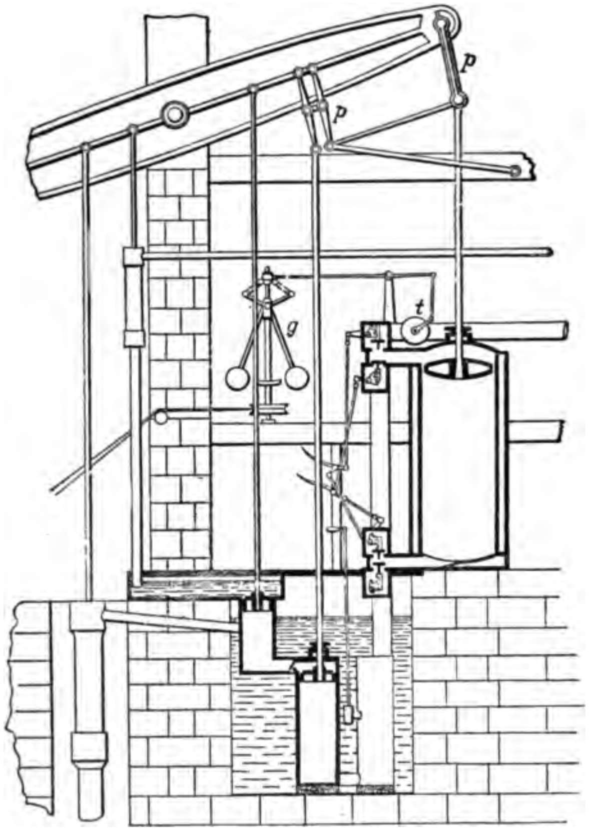


Fig. 4.

Watt also invented the "indicator," by means of which diagrams are made which show at all points the relation between the pressure in the cylinder and the position of the piston at that instant. His assistant, Murdoch, invented the slide valve as a means of admitting and releasing the steam. Fig. 4 shows Watt's final engine.

Watt, like other early inventors, sold many of his engines to miners, who had been using horses to pump out the mines; and for this reason he rated his engines by the horse-power. Although this term has an historical derivation it has no real significance, and no relation whatever to the power of a horse. It is an established unit for measuring the rate at which work is done. One horse-power is the amount of work necessary to raise 33,000 pounds through one foot in one minute; and we may say that one horse-power is equal to 33,000 foot pounds per minute.

Watt saw that by using high-pressure steam he could get more work from it; but as it was not possible to make very reliable boilers he never used a pressure of more than seven pounds per square inch above the atmosphere. About the year 1800 comparatively high pressures came more into use and the *non-condensing* engine was introduced. In Watt's engine, and all those preceding his, a vacuum was produced in front of the piston by condensing the steam, and either the atmosphere or steam at atmospheric pressure pushed it through the stroke. In the non-condensing engine, using high-pressure steam, the space in front of the piston could be opened to the atmosphere at exhaust, and although the atmospheric pressure resisted its motion the pressure of the steam behind the piston was still greater than that of the air. These engines were much more simple than the condensing engines, as they required no condenser.

About this time what would now be called a compound engine was introduced by Hornblower and later by Woolf. It had two cylinders of different size. Steam was admitted into the smaller cylinder, and then passed over into the larger. The steam expanded a little in the smaller cylinder and much more in the larger one.

A great many attempts were made to build *locomotives*, but they were generally unsuccessful until George Stephenson built his engine, the "Rocket," in 1829. The principal new feature of this engine was the improved *steam blast* for increasing the draft in the furnace and so making possible the use of a smaller boiler. Later he used the "link motion," which enabled the engine to be quickly reversed and the amount of expansion varied.

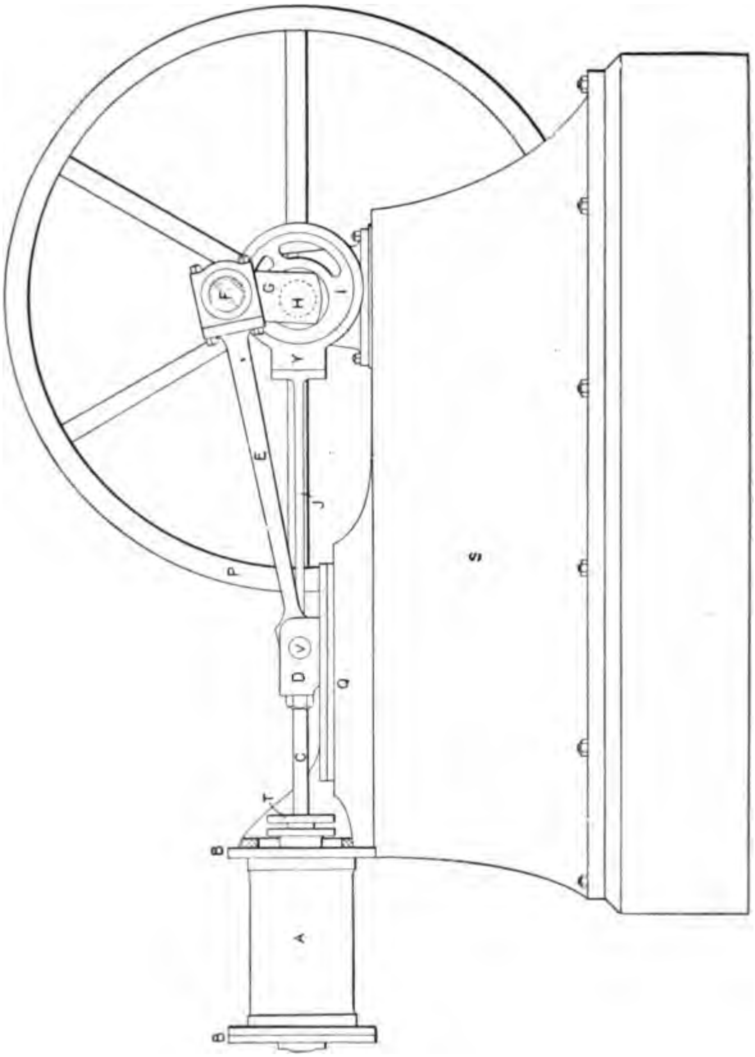


Fig. 5.

The Stephenson link motion may be seen on almost any locomotive. It is simply a device by which either of two eccentrics may be made to move the valve.

About the year 1814, Woolf introduced a compound pumping engine in the mines of Cornwall, but a simpler engine was later introduced and Woolf's engine fell into disuse. This later engine became known as the Cornish Pumping Engine and was famous for many years because of its economy. It was the first engine ever built that could compare at all with modern engines in the matter of steam consumption. It consisted of a single cylinder placed under one end of a beam from the other end of which hung a heavy rod which operated a pump at the foot of the shaft. Steam was admitted on the upper side of the piston for a short portion of the stroke and allowed to expand for the remainder of the stroke. This forced the piston down, lifted the heavy pump rod and filled the pumps with water. Then communication was established between the upper and under side of the piston, exhaust occurred, and the heavy pump rod fell, lifting the piston and forcing the water out of the pumps. The cut-off was about .3 stroke, and the pump made about seven or eight complete strokes per minute with a short pause at the end of each stroke to allow the valves to close easily and the pumps to fill with water. The cylinder was jacketed. These engines needed great care and were in charge of competent men, to whom prizes were frequently given for the best efficiency, which doubtless accounts for their wonderful performance.

PARTS OF THE STEAM ENGINE.

Fig. 5 shows the elevation of a simple form of steam engine.

The Cylinder A (see plan, Fig. 6) is that part of the engine in which the piston moves back and forth. It is made of cast iron and accurately bored. Great care must be taken in this work, for any unevenness will allow steam to leak through between the piston and cylinder walls or it may even cause the piston to stick or work hard. In large engines the cylinder consists of two parts, the outer or cylinder proper and a comparatively thin cast iron liner. A space can be left between them for a steam jacket. Should the cylinder liner be damaged, it can be replaced without the expense of a new cylinder.

The Cylinder Heads B cover the ends of the cylinder and are securely bolted thereto. In the crank-end cover there is an opening for the piston rod to pass through. This opening is made steam tight by a stuffing box which surrounds the piston rod. Sometimes the piston rod is prolonged beyond the piston and through the front cover. This extension of the piston rod is to help steady the piston in a long stroke and is known as the tail rod. When a tail rod is used another stuffing box must also be provided for the head-end cover.

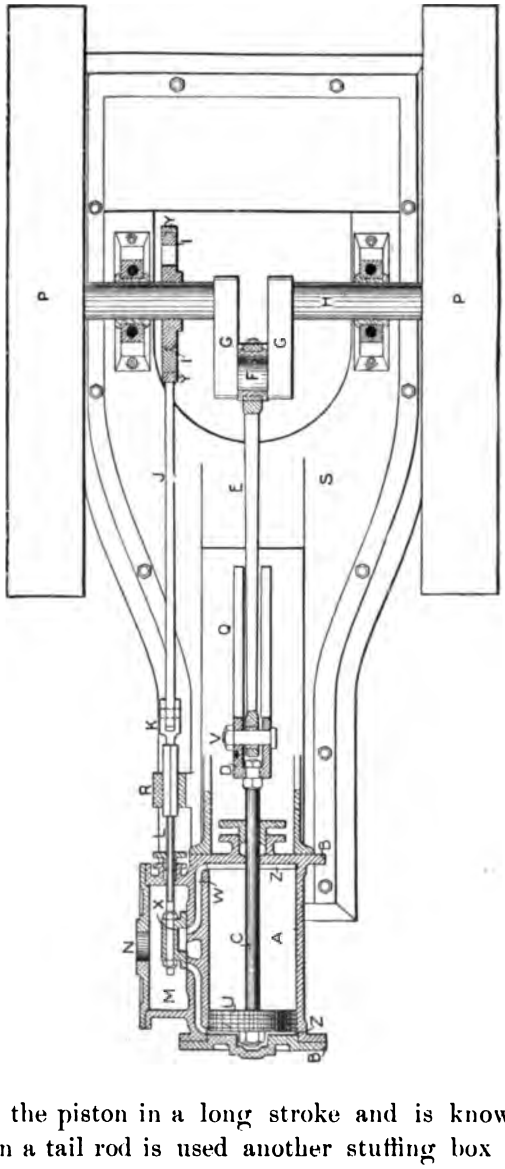


Fig. 6.

The Piston U, in small engines, is usually a thick disc of iron or steel, as shown in Fig. 6. It is often made conical, as shown in Fig. 7, to better withstand the steam pressure and to gain space. The piston must fit the cylinder steam tight and yet move easily. To accomplish this, one or more grooves in the piston are filled with packing (usually metallic), or spring rings may be used.

The Piston Rod C (Figs. 5 and 6) is made of steel and connects the crosshead and the piston to which it is rigidly fixed.

The Crosshead D serves to join the piston rod and connecting rod. At one end it is fastened to the piston rod, and at the other end is the wrist pin V on which the connecting rod swings. It is guided to and fro by the crosshead guides Q.

The Connecting Rod E is a steel forging from three to eight times the length of the crank, depending upon the type of engine. One end is jointed to the crosshead by the pin V, called the wrist pin, while the other encircles the crank pin and revolves with it. A detail view of one end is shown in Fig. 8; the other end is frequently similar. In some cases the small end is forked, as shown in Fig. 9.

The Crank Pin F forms the connection between the crank and connecting rod.

The Crank G, equal in length to one-half the stroke of the piston, converts the back and forth motion of the connecting rod into circular motion. It may be simply an arm, as shown in Fig. 10, or a complete disc keyed to one end of the shaft, as shown in Fig. 11. The disc is more nearly balanced than the crank.

The Shaft H transmits the rotary motion from the crank to the fly wheel P.

The Frame of the engine S is a heavy casting, which supports the cylinder and bearings. It should be securely bolted to the foundation.

The Steam Chest M receives steam directly from the boiler, and the steam passes thence through the ports W into the cylinder.

The Eccentric I is a disc keyed to the shaft so that its center and the center of the shaft do not coincide. The eccentric strap Y encircles the eccentric and imparts a reciprocating motion to the valve stem L and the eccentric rod J. This action is similar to that of the crank and connecting rod, but exactly reversed. K is the valve stem crosshead and R its guides.

The Slide Valve X is the valve for alternately admitting

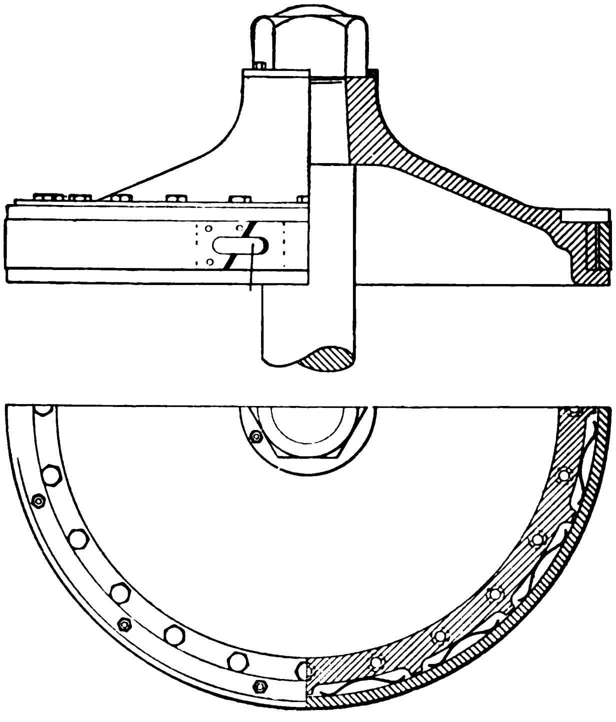


Fig. 7.

steam to the cylinder and releasing it. It has a cup-shaped cavity in its face through which the exhaust steam passes. It is situated in the steam chest and is moved by the valve gear, that is, the eccentric and the eccentric rod.

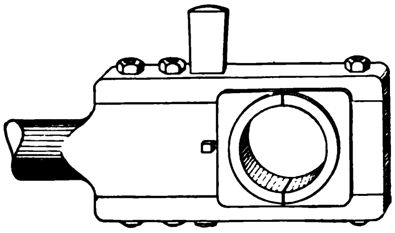


Fig. 8.

The Clearance Z is the space between the piston and the cylinder head (when the piston is at the end of the stroke), together with the volume of the steam ports. This volume must be filled with steam before the piston can start. It is usual to express the clearance as a certain per cent of the volume swept through by the piston.

The crank may revolve in either direction. If we stand by the cylinder, facing the crank shaft, and the crank moves away from us as it passes over the shaft, we say that it is running *over*. If it moves away from us as it passes under the shaft, we say that the engine is running *under*. The action of steam in the cylinder of an engine is very complicated, and its discussion will be taken up later in the course.

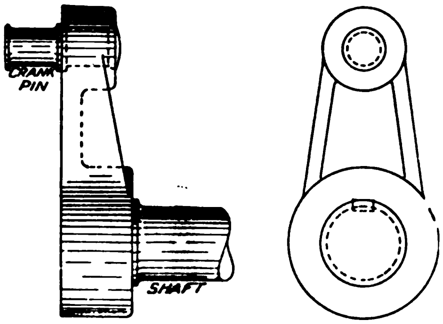


Fig. 10.

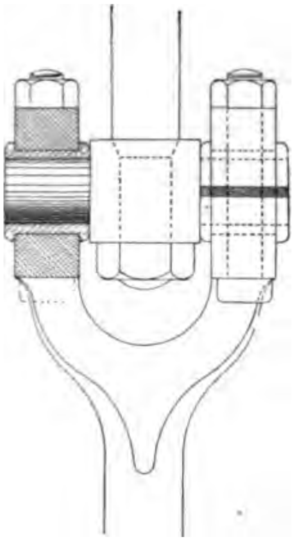


Fig. 9.

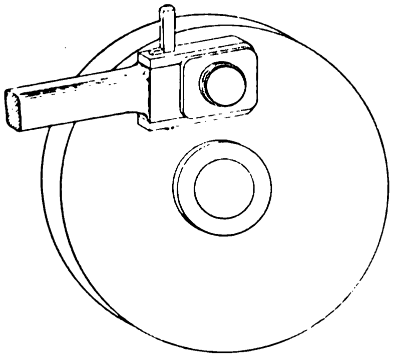


Fig. 11.

TYPES OF ENGINES.

Classification. There are so many different types of engines that it is difficult to classify them all properly. Most engines belong to several classes at one and the same time. For instance, there are condensing and non-condensing engines; there are simple, compound, triple and quadruple expansion; there are high speed and low speed, vertical and horizontal, locomotive, stationary

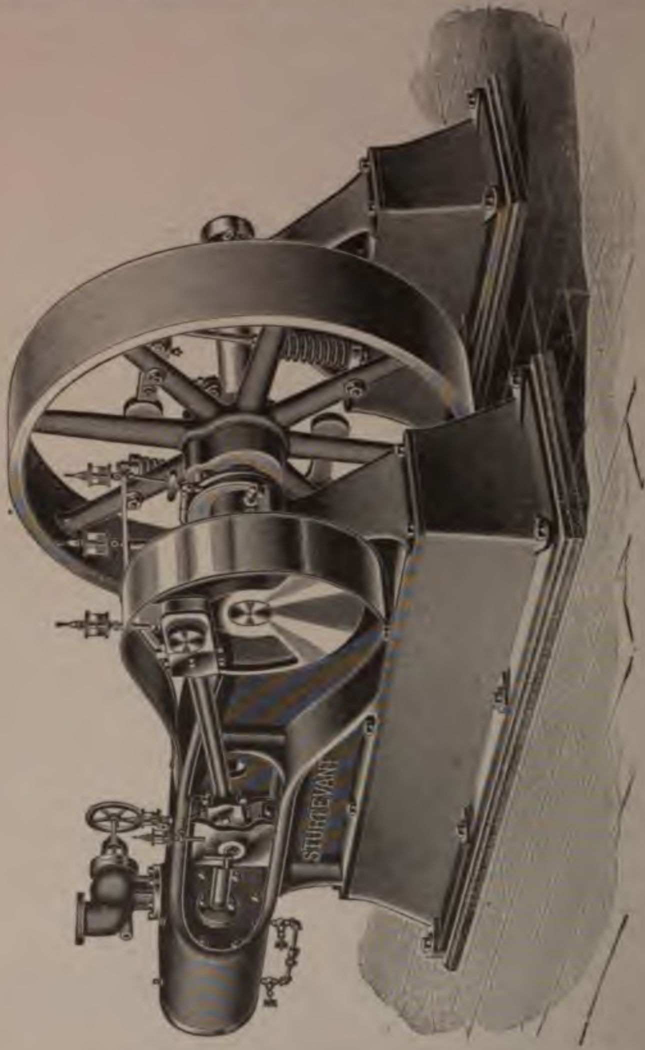


Fig. 12.

and marine, and many other classes into which these might be further subdivided.

Simple Engines. The simplest type of engine is the simple expansion. It has one cylinder and admits steam for a part of the stroke, expands it during the remainder and exhausts either into the atmosphere or into a condenser. Simple engines (see Figs. 5 and 12) are now used only for comparatively small powers, say 100 H. P. or less, and although more extravagant of fuel than the others, may still be the most economical financially if low first cost is an important item, if they are not run continuously, or if the load fluctuates widely.

Compound Engines have two cylinders known as the high pressure and low pressure. Steam enters the smaller or high pressure cylinder and then expands until release, when it is exhausted into the larger cylinder, where the expansion is finished. The cylinders should be so proportioned that approximately the same amount of work can be done in each. The first cylinder is small, because it has the higher steam pressure, and a given weight of steam occupies less space when at high pressure. The second must be large, so that the volume at cut-off can contain all of the steam exhausted from the high.

Besides being more economical the compound has a distinct mechanical advantage. The two cranks may be set at right angles, so that when one is on dead center the other is at a position of nearly its greatest effort. This makes a dead center impossible, and gives a more uniform turning moment. Then the individual parts may be made lighter, and are thus more easily handled, but the engine is much more costly, and it is nearly twice as much work to take care of it.

When the cranks of a compound engine are at 90°, the low-pressure piston is not ready to receive steam when the high pressure exhausts, hence there must be a receiver to hold the steam until admission occurs in the low. Such engines are called *cross compound.* Fig. 13 shows one form. Sometimes instead of having the cranks at 90° they are placed together or opposite. Then the strokes begin and end together, and the high can exhaust directly into the low without a receiver. Such engines are called *Woolf engines.* A tandem compound engine, shown in Fig. 14, has both pistons on one rod, the high-pressure piston rod forming the low-pressure tail rod. Such engines are less expensive because there is but one set of reciprocating parts instead of two, but like simple engines they have the disadvantage of dead points.

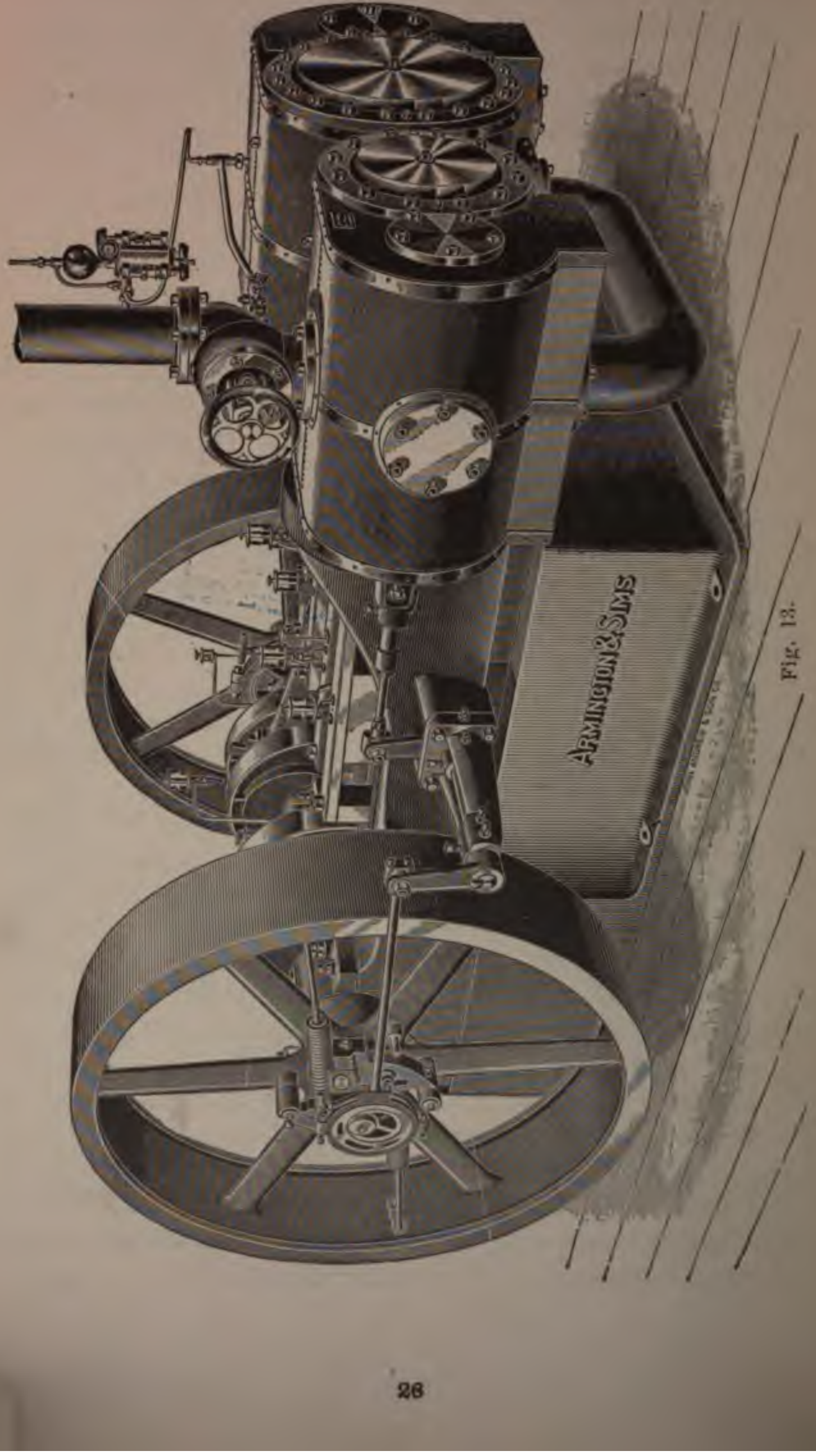


Fig. 13.

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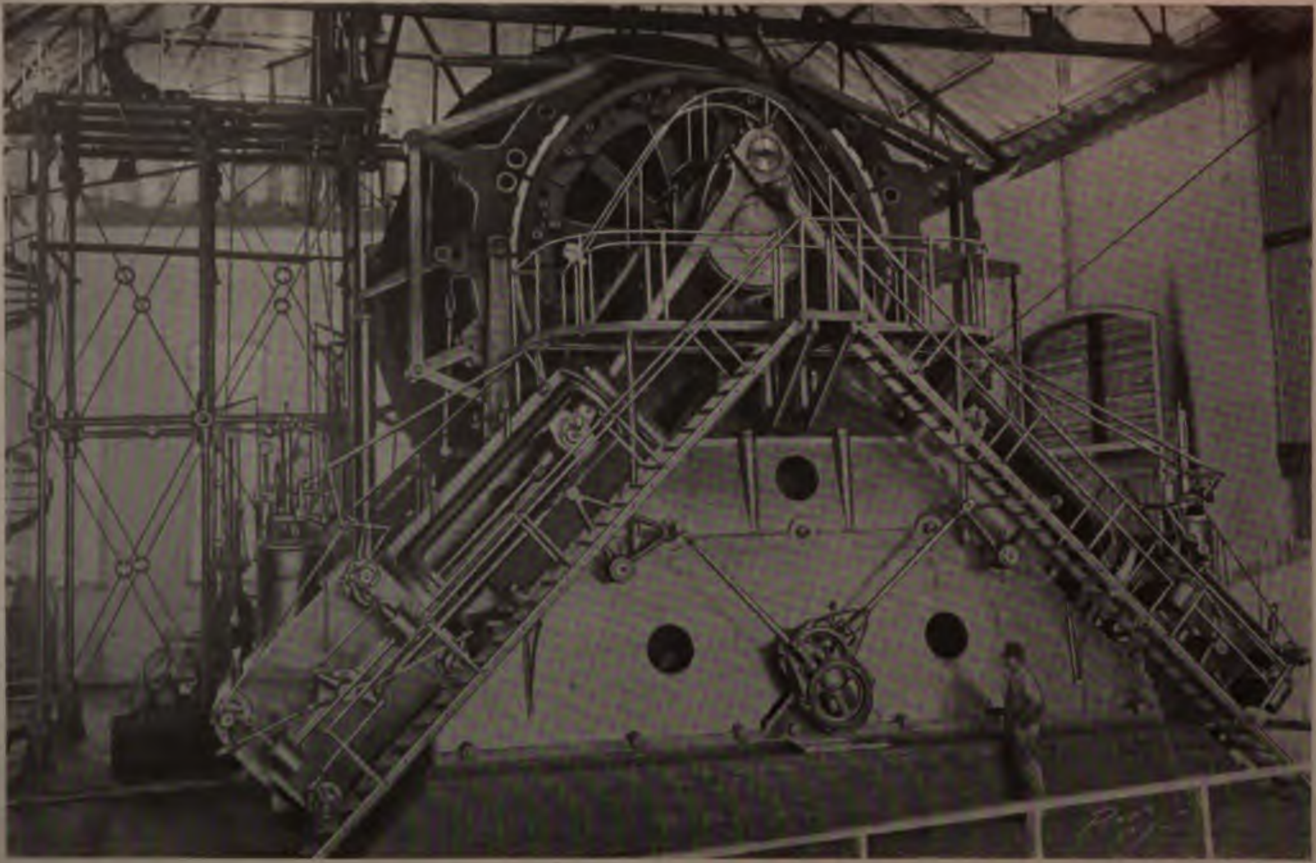
ASTOR, LENOX
TILDEN FOUNDATIONS



TAMARAK HOISTING ENGINE AT MINE.

Nordberg Manufacturing Co.

Triple Expansion Engines expand the steam in three stages instead of two. There are usually three cylinders, the high, intermediate, and low, arranged with cranks 120° apart. This gives a more uniform turning moment than a compound. Sometimes there are four cylinders to the triple, one high, one intermediate, and two low. This arrangement gives better balance and is often used in marine work.

For triple engines there must be a receiver between each two cylinders. Fig. 15 shows the essential features of a triple expansion engine.

Quadruple Engines expand their steam in four stages instead of three. Multiple-expansion engines are nearly always condensing.

Cylinder Ratios. There are several considerations to be remembered when proportioning the cylinders of multiple-expansion engines. The ratio of the cylinders should be such that each develops nearly the same power; the drop in pressure between the cylinders and receivers should be small, and the strains in the cylinders about equal.

There are many formulas in use, some simple, others involving mathematical calculation. A common rule for compound engines is to make the ratio of the cylinders equal to the square root of the total ratio of expansion. Thus if the steam has a ratio of expansion of 9, the ratio of the cylinder volumes will be $\sqrt{9} = 3$, or the low-pressure cylinder will have a volume 3 times as great as the high-pressure cylinder. If the cylinder ratio is 3, and the length of stroke is the same for both, the diameter of the low-pressure cylinder will be 1.75 times that of the high-pressure cylinder.

Another rule is to make the cylinder ratio equal to the total ratio of expansion multiplied by the fractional part of the stroke completed when cut-off occurs in the high-pressure cylinder.

Suppose the ratio of expansion is 9, as above, and that cut-off occurs at $\frac{1}{3}$ of the stroke in the high-pressure cylinder. The ratio of cylinder volumes will be $9 \times \frac{1}{3} = 3$. If cut-off occurs at $\frac{1}{2}$ the stroke, the ratio will be $9 \times \frac{1}{2} = 4.5$.

For triple expansion engines the low pressure cylinder is made large enough to develop the whole power if steam at boiler pressure is used.

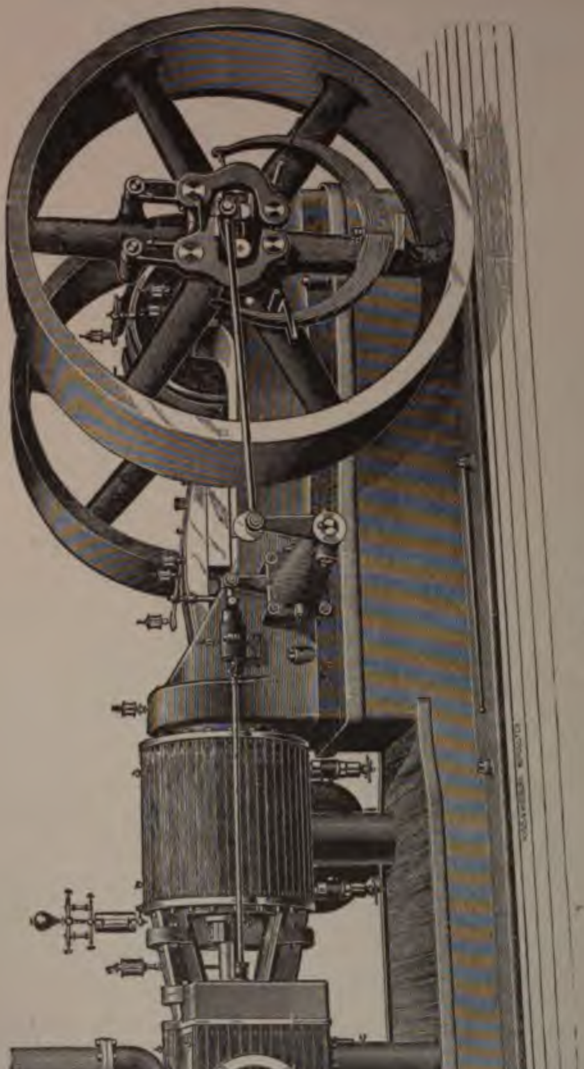


Fig. 14.

The intermediate cylinder is made approximately of a mean size between the high and the low. The area of the intermediate piston is found by dividing the area of the low by 1.1 times the square root of the ratio of the low to the high.

We may write the above thus,

$$\text{Area of high-pres. cyl.} = \frac{\text{Area of low-pres. cyl.}}{\text{Cut-off of high-pres.} \times \text{ratio of exp.}}$$

$$\text{Area of inter. cyl.} = \frac{\text{Area of low-pressure cylinder.}}{1.1 \times \sqrt{\text{ratio of low to high.}}}$$

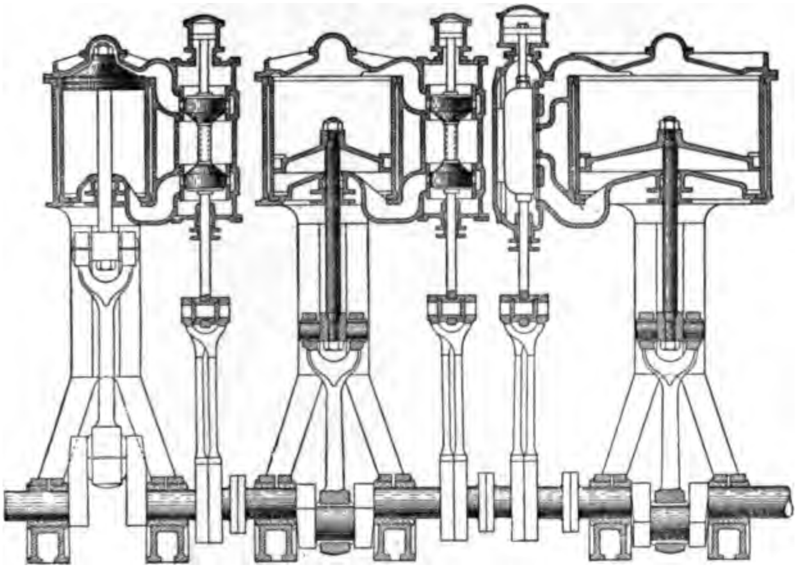


Fig. 15.

In general the volumes for triple expansion are of about the following ratios, 1 representing the volume of the high pressure cylinder:

1 : 2.25 to 2.75 : 5 to 8.

For quadruple-expansion engines the ratios are as follows:

1 : 2 to 2.33 : 4 to 5 : 7 to 12.

High Speed Engines. Of late years there has been a demand for engines of higher speeds than were formerly used. It was found desirable to run dynamo-electric machines by connecting them ***directly*** to the shaft of the engine rather than by belts as before.

This required engines running from 200 to 1,000 revolutions per minute instead of from 60 to 90 revolutions. Also for engines in torpedo-boats, speeds as high as 400 or 500 revolutions are common.

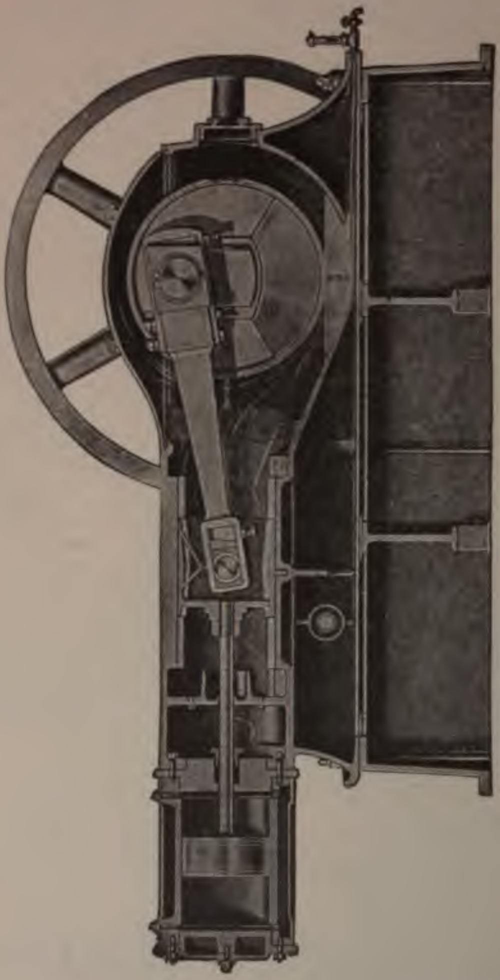


Fig. 16.

Running at high speed requires various changes. Reciprocating parts must be made lighter to reduce the vibration, and must be more carefully proportioned to maintain balance. Bearings must be made very much larger to reduce the pressure in order that the friction may not be excessive and cause heating. Special care is necessary that bearings should be tight, since the least looseness will cause knocking and hammering until the bearing is ruined. Parts must be as simple as possible and so arranged as to need the least possible attention. In slow-speed engines the engineer can watch the oil cups and oil any part while it is running. But this is impossible in high-speed engines, and special means must be used to insure plenty of oil to the bearings and the cylinder.

These peculiarities of high-speed engines may be easily seen in any engine for electric lighting, for running fans, etc. It is also necessary that the speed of engines for running electric generators should be very steady, as the slightest change in the speed make the lights flicker. The fly wheels are, therefore, larger or heavier than for other types and the governors are made specially sensitive. Fig. 16 shows a horizontal high-speed engine with the working parts encased. Fig. 17 shows a vertical high speed.

For any *double-acting* engine, that is, for any engine in which the steam acts first on one side and then on the other side of the piston, the piston first pushes and then pulls on the connecting rod and crank. At each half revolution of the crank the direction of the pressure reverses. It is this change of pressure which causes the pounding if the bearings are at all loose. This is one of the greatest troubles with high-speed engines. In order to avoid these rapid reversals in pressure, *single-acting* vertical engines are used to a considerable extent. In such engines the steam is admitted only to the *head end* of the cylinder. The other end stands open. The connecting rod is in compression throughout the whole revolution instead of being first in compression and then in tension. Besides insuring that the piston shall always push, this arrangement simplifies the valves.

There are several good single-acting, high-speed engines. One of the earliest made was introduced by Brotherhood. He used three cylinders, set around the shaft 120° apart. Another well-known example is the Willans "central valve" engine. These are both English engines. A well-known American engine of this type is the Westinghouse high-speed engine, a section of which is shown in Fig. 18.

Vertical and Horizontal Engines. At the present time the most common type of engine is the horizontal direct-acting, that is, an engine whose cylinder is horizontal and whose piston acts on the crank through a piston rod and connecting rod. In small engines the whole is often on one bed plate. Such engines are called self contained. The cylinder is either bolted to the back of the bed plate or rests directly on it.

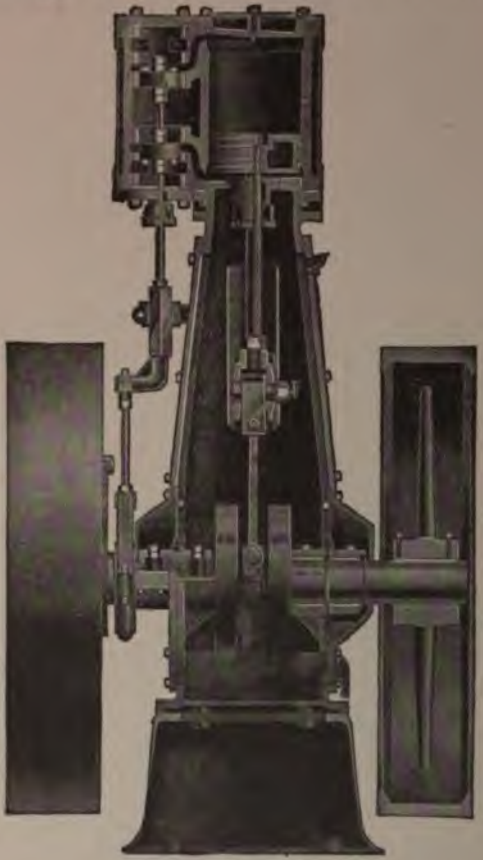


Fig. 17.

In marine work vertical engines are used in almost every case. The reason for this is, of course, the saving of floor space, which is so important in a vessel. This saving of space, however, is also very important in many cases such as in crowded engine rooms of cities where land is expensive, and as there are a number of advantages which vertical engines have over horizontal, they are coming largely into use in stationary practice.

A second advantage of the vertical over the horizontal engine is the reduction of cylinder friction and unequal wear in the cylinder of the latter. In the horizontal engine the piston is generally supported by resting on the cylinder, which is gradually worn until it is no longer round. This causes leakage of steam from one side to the other. Evidently this is entirely avoided in the vertical engine.

Still another advantage of the vertical engine is the greater ease of balancing the moving parts so that there shall be no jarring or shaking. It is impossible to perfectly balance a steam engine of one or two cylinders. If it is balanced so that there is no tendency to shake sidewise, it will shake endwise; and if it is balanced endwise it will shake sidewise. The jarring is due to the back and forth motion of the reciprocating parts and the centrifugal force of the crank and connecting rod. The crank can be readily balanced by making it extend as far on one side of the shaft as it does on the other, but the piston and connecting rod are more difficult to balance. We can greatly reduce the effect of jarring if we balance the crank and make the endwise throw come in line with the foundation, which should be heavy enough to absorb the vibration transmitted. In a horizontal engine this endwise throw not being in line with the foundation will cause vibration in the engine itself.

In machines that can be anchored down to a massive foundation a state of defective balance only results in straining the parts and causing needless wear and friction at the crank-shaft bearings and elsewhere, and in communicating some tremor to the ground. The problem of balancing is of much more consequence in locomotive and marine engines.

To sum up the general advantages of vertical engines: they have less cylinder wear, they take up less floor space, and they can be better balanced. In addition to these there are certain advantages which vertical engines have for certain kinds of work.

The disadvantages of vertical engines are as follows: The pressure on the crank-pin is greater during the *down* stroke than during the *up* stroke because, during the down stroke the weight of the reciprocating parts is added to the steam pressure, and during the up stroke this weight is subtracted.

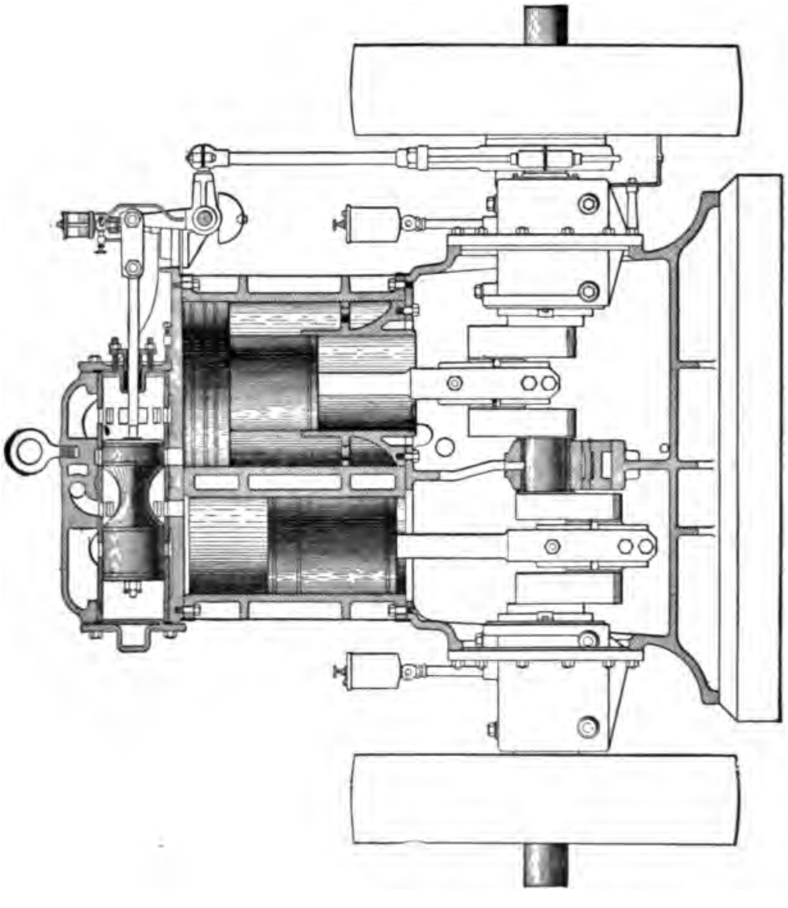


Fig. 18.

Another difficulty is that in large engines the various parts are on such different levels that they require considerable climbing. This requires more attendants and is sometimes the cause for neglect of the engine.

The foundations for vertical engines generally need to be *deeper* than those for horizontal engines. At the same time, however, they need not be as *broad*.

Marine Engines. The first steam vessels were fitted with paddle-wheels, and as beam engines were the most common, this form of engine was used. Its construction, however, was somewhat modified for this service. This arrangement of beam engine and paddle-wheel was used for many years and was applied to ocean vessels as well as to small river boats. It is still used, especially in this country, on river steamers and some coast steamers. The beam is supported by large A frames on the deck, and the engines are about on a level with the shaft.

Engines of this type take up rather more room than those now in common use, partly because of great size, and also because of the shaft and paddle-wheels. Another disadvantage is that in heavy weather, when one paddle-wheel is thrown out of water the other is deeply immersed and takes all the strain, so that there is a tendency to rack the boat. Then again if the boat is loaded heavily the paddle-blades are very deeply immersed while if light they barely touch the water. It is hard to handle the engines satisfactorily under both conditions.

The introduction of the screw propeller overcame these difficulties very largely and at the same time required a much faster running engine. At first, the increased speed was supplied by the use of spur-wheel gearing, but gradually higher speed engines were built and connected directly to the propeller shaft. It was, of course, difficult with small width at each side of the shaft to use horizontal engines, therefore various arrangements of inclined engines were used before the vertical engine was finally chosen by all as the standard form for marine work. It is only in recent years that the vertical engine has become general in naval work and in merchant steamers.

In merchant ocean steamers the common form has three cylinders set in line, fore and aft, above the shaft. The cranks are 120° apart, to give a very even turning moment. The three cylinders are worked *triple expansion.* The valves are usually piston valves on the high and intermediate, and double ported slide valves on the low. Sometimes piston valves are used on all the cylinders. Plain slide valves are not suitable for high-pressure work of any kind.

For engines on ocean vessels it is necessary to use surface condensers in order that the same water may be used over and over again. If it were necessary to take in sea-water for the boilers they would soon become clogged with the salt and require cleaning. Surface condensers for marine work are generally made up of a large number of brass tubes of from $\frac{3}{4}$ inch to 1 inch in diameter. In some cases the cold water is forced to flow through the tubes while the steam comes in contact with the outside of the tubes.

In any marine plant there are four special pumps. The ***first*** is the air pump for the condenser. This is usually made large so that in case there is a leak in the condenser it can take charge of the water even if it becomes necessary to run as a jet instead of a surface condenser. The *second* is the feed pump for the boilers. The *third* is the circulating pump, which forces the current of cold water through the condenser. The last is the bilge pump, which pumps out water that gathers in the bilge of the ship by leakage or otherwise. In case of a serious leak all the pumps can be made to pump from the bilges. In some old types all these pumps were worked from the main engine; generally, however, the feed pump and the circulating pump are separate, as also the bilge pump. The circulating pump is, in many modern engine rooms, of the centrifugal type.

Locomotive Engines. Of all steam engines the most inefficient is the steam locomotive. In the first place, the boiler must be forced so hard that the products of combustion pass off at a very high temperature and consequently carry away a great deal of heat. Bits of entirely unburned or partially burned coal are drawn through and wasted.

In the second place, the boiler is exposed to great loss of heat by radiation. Although its surface is lagged, it cannot be very effectively covered, and the wind takes away a great deal of heat.

Mechanically also the locomotive is very imperfect. In most good steam engines the efficiency, that is to say, the ratio of the *effective horse-power* to the *developed horse-power* is fully $\frac{9}{10}$ or 90 per cent. In the locomotive this ratio was shown to be about 43 per cent by two independent tests. This is in part due to the special difficulties in locomotive construction, but the principal losses are those caused by radiation and the escape of heat from the stack.

As to locomotive boilers, Mr. Forney says, "The weight and dimensions of locomotive boilers are in nearly all cases determined by the limits of weight and space to which they are necessarily confined." It may be stated generally that within these limits a locomotive boiler cannot be made too large. In other words, boilers for locomotives should always be made as large as possible under the conditions that determine the weight and dimensions of the locomotives.

There are certain types of locomotives common in American practice which have special names. The eight-wheel or "American" passenger type of locomotive has four coupled driving-wheels and a four-wheeled truck in front. The "ten-wheel" type has six coupled drivers and a leading four-wheel truck. This type is used for both freight and passenger service. The "Mogul" type is used altogether for freight service; it has six coupled drivers and a two-wheel or pony truck in front. The "Consolidation" type is used for heavy freight service. It has eight coupled driving-wheels and a pony truck in front. There are also a great many special types for special purposes. In switch-yards a type of engine is used which has four or six drivers with no truck. The Forney type has four coupled driving-wheels under the engine and a four-wheel truck carrying the water-tank and fuel. This type is used on elevated roads largely. "Decapod" engines are a type used for heavy freight service, having ten coupled driving-wheels and a two-wheel truck in front. A tank engine is one which carries the feed water in tanks on the engine itself instead of in the tender, as in other engines. The various different forms are too numerous even to name.

There has been some effort made to introduce compounding in locomotive practice. It has in some cases been very successful, especially for express trains. A committee of the American Railway Master Mechanics Association says of compounding:

"(*a*) It has achieved a saving in the fuel burnt, averaging 18 per cent at reasonable boiler pressures.

(*b*) It has lessened the amount of water to be handle.

(*c*) The tender can, therefore, be reduced in size and weight.

(*d*) It has increased the possibilities of speed beyond sixty miles per hour, without unduly straining the engine.

(*e*) It has increased the haulage power at full speed.

(*f*) In some classes of engines it has increased the starting power.

(*g*) It has lessened the valve friction per horse-power developed."

A number of other reasons are given in their report.

In opposition to this may be mentioned the extra first cost of the engine and the cost of maintaining a more complicated machine. It is much more work to keep it in repair and many engineers are of the opinion that the saving in fuel is only sufficient to offset these extra expenses. If the engine is running under steady load, the compounding will effect a great saving; but in many parts of the country the load varies constantly, due to grades in the road.

We shall learn later that a compound engine cannot work efficiently under light load. If the grades are first up and then down, the simple engine is the more economical. For a steady upgrade the compound is more economical, as it can be run steadily under full load. This is especially true in mountain districts, where the long up-grades and scarcity of fuel and water make ideal conditions for compound locomotives. Through freight service probably offers the widest field for success with these engines.

It has already been said that it is difficult to balance an engine completely. This defect is very much greater in locomotives than in stationary engines. Lack of balance in a locomotive results in serious pounding of the track. Also there is danger of flattening and breaking the wheels, and the rails may be seriously damaged.

Pumping Engines. The first steam engines built were pumping engines and today the most economical engines are those built for this work.

In pumping engines it is not absolutely necessary to have a revolving shaft. All that is required is that the piston in the pump cylinder shall be driven back and forth with a plain reciprocating motion which may be exactly like that of the steam piston. For this reason, in early pumping engines and also in many modern engines, the reciprocating motion of the steam piston is applied directly, or through a beam, to produce the reciprocating motion of the pump piston or plunger without the use of any revolving part. Frequently, however, it is desirable to use a fly wheel so that the steam may be used expansively, and in these cases, of course, a revolving shaft must be used. Fig. 19 shows a power pump.

For deep-well or mine pumping, the cylinders are often set in a vertical position directly over the pump cylinder. The piston rod extends from the steam cylinder directly below to the pump plunger. Sometimes it is possible to use steam expansively in these pumps by reason of the weight of the reciprocating parts. When the weight is sufficient, the steam can be cut off before the end of the stroke and the momentum of the parts will be enough to just finish the stroke, consequently these pumps are sometimes compounded. They are possible only in pumping from very deep wells.

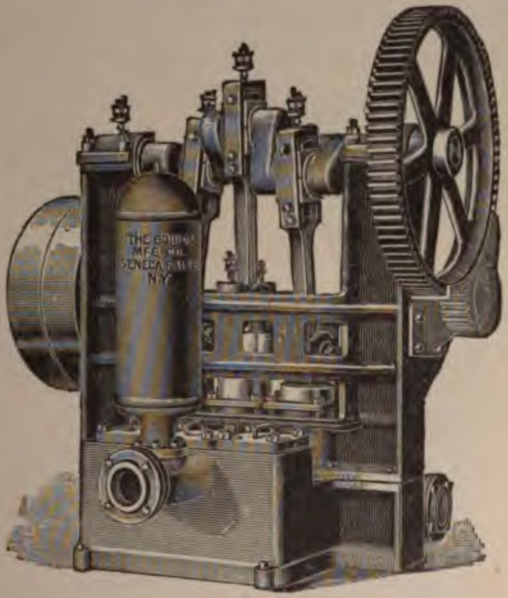


Fig 19.

Direct-acting Steam Pumps. Fly wheel pumps have one disadvantage, if run too slowly the momentum of the fly wheel is not sufficient to carry it by the dead centers; if run too fast the fly wheel is in danger of bursting. A fly-wheel pump can be made to discharge a small amount of water by means of a by-pass valve, but of course it then runs at a disadvantage.

The direct-acting steam pump shown in Fig. 20 has the steam piston at one end of a rod and the water piston at the other end. The steam pressure acts directly on the piston; no fly wheel is used, and since the reciprocating parts are comparatively light, and there is no revolving mass to carry by the dead points, it is evident that in the ordinary form there can be no expansion of steam. The pump is inexpensive and gives a positive action. It uses a great quantity of steam relatively, but for small work the absolute amount is not very great. Even in larger engines the lighter foundations that are possible and the slight first cost are frequently controlling features.

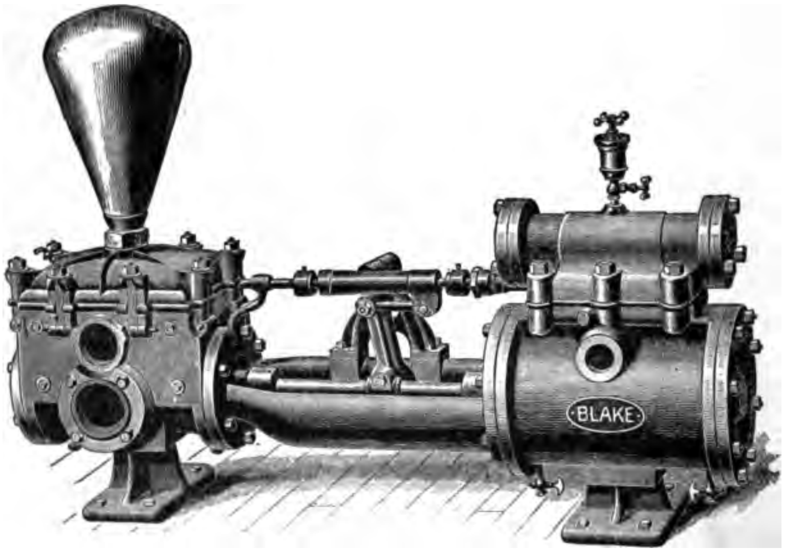


Fig. 20.

A rocker or bell-crank lever on the piston rod moves the steam valve and admits steam to the other side of the piston while opening the first side to exhaust. In large pumps of this kind, and even in some small ones, this motion merely admits steam to a small auxiliary piston which then moves the main steam valve by steam pressure. Some pumps operate the steam valve by means of a tappet instead of a rocker and bell-crank lever.

There have been various devices tried for using steam expansively in these direct-acting pumps without the use of a fly-wheel.

In order to do this it is necessary to provide some means of storing energy during the early part of the stroke and returning it during the latter part, when, owing to the expansion, the pressure of the steam is less. One such device is as follows: a crosshead A, (Fig. 21) fixed to the piston rod is connected to the plungers of a pair of oscillating cylinders B B, which contain water and communicate with a reservoir full of air compressed to about 300 pounds per square inch. When the stroke (which takes place in the direction of the arrow) begins, these plungers are first forced in, and hence work is at first done on the main piston rod, through the compensating cylinders B B, on the compressed air in the reservoir. This continues until the crosshead has advanced so that the oscillating cylinders stand at right angles to the line of stroke. Then for the remainder of the stroke their plungers assist in driving the main piston, and the compressed air gives out the energy which it stored in the earlier portion.

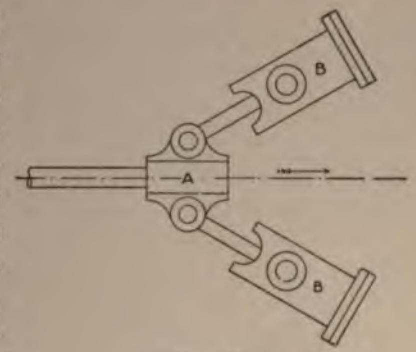


Fig. 21.

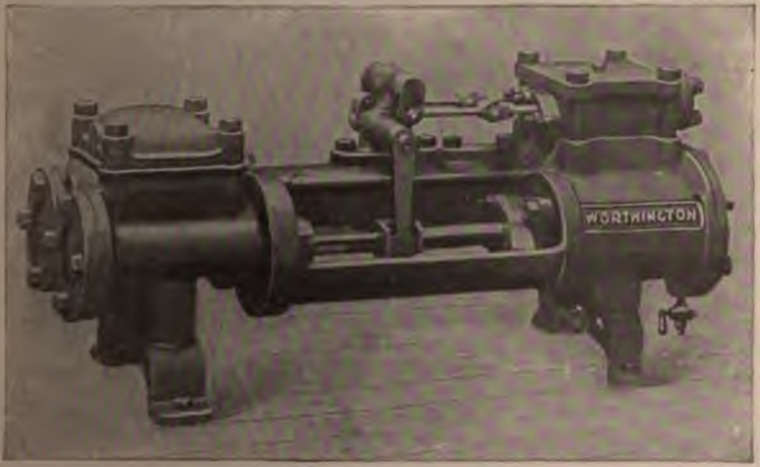


Fig. 22.

The Duplex steam pump consists simply of two direct-acting steam pumps placed side by side, as shown in Fig. 22. On the piston rod on one side is a bell-crank lever which operates the valve of the other pump. On the further piston rod is a rocker arm which moves the valves of the first pump. There must be a rocker on one side and a bell-crank lever on the other because of the relative motion of the valves and pistons. The first piston, as it goes forward, must use a rocker because it draws the second valve back. The second piston, as it goes back, must use a bell-crank lever because it must push the first valve back in the same direction as its own motion. The two pistons are made to work a half-stroke apart. Thus one begins its stroke when the other is in the middle. In this way a steadier flow of water is obtained, for both pumps discharge into the same delivery pipe. The pumps may be made compound. A sectional view of the pump is shown in Fig. 23.

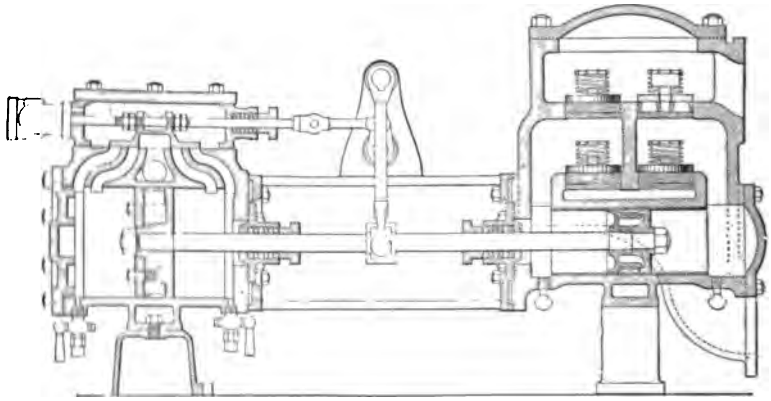
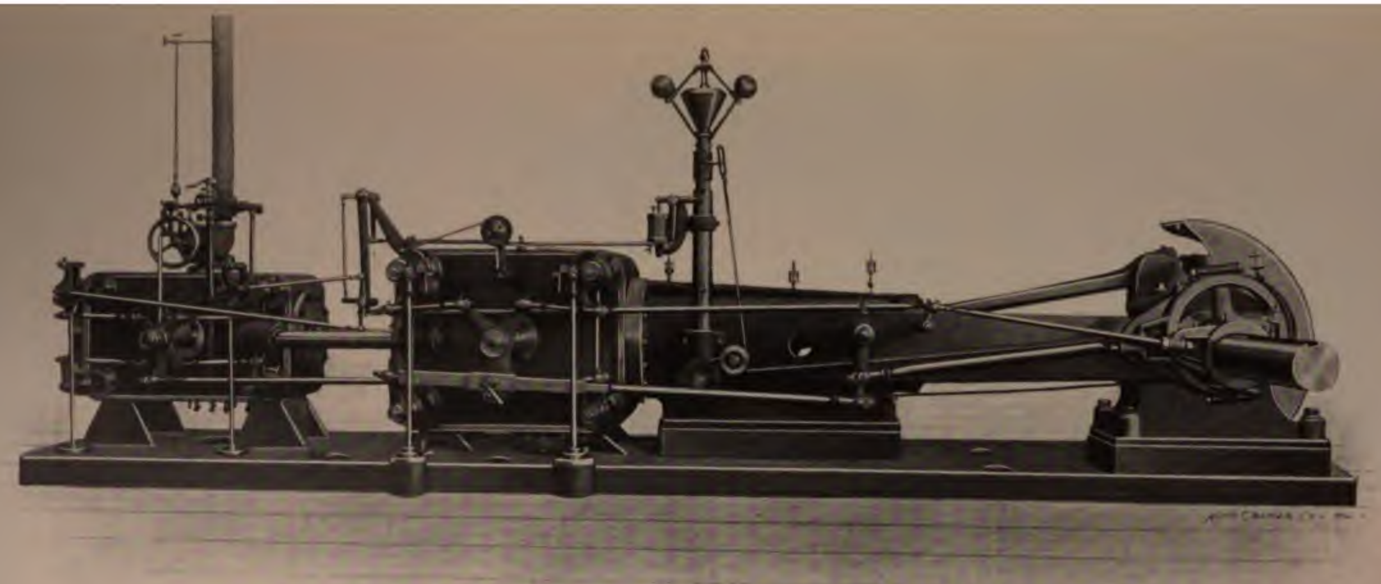


Fig. 23.

Corliss Engines. In large engines a common way of regulating the steam supply is by changing automatically the point in the stroke of the piston at which the steam is cut off. This is frequently accomplished by using some trip gear similar to the one first introduced by Geo. Corliss. These gears are called Corliss gears. In the Corliss gear there is a separate admission valve and a separate exhaust valve for each end of the cylinder, as shown in Fig. 24. The exhaust valves are opened and closed by



TANDEM CORLISS ENGINE DRIVING ELECTRIC LIGHT PLANT. TRADESMEN'S NATIONAL BANK, PITTSBURG.
Nordberg Manufacturing Company.

THE NE
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TILDEN

the motion of rods or cranks connected to them. The admission valves are opened in the same way, but they snap shut by themselves when the piston has reached a certain point of its stroke. This point will vary with the position of the governor, which in turn depends on the speed of the engine. These Corliss engines cannot be run at high speed because the trip gear requires some time to act.

The valves of Corliss engines turn in hollow cylindrical seats which extend across the cylinders. A wrist plate which turns on a pin on the outside of the cylinder receives a motion of oscillation from an eccentric and opens the valves by means of the rod connections. When the piston has reached a point where the steam should be cut off, the trip gear is held in such a position by the governor that it releases the valve, which now springs shut under the action of the dash-pot. The admission valve to the other side of the cylinder is controlled in exactly the same way.

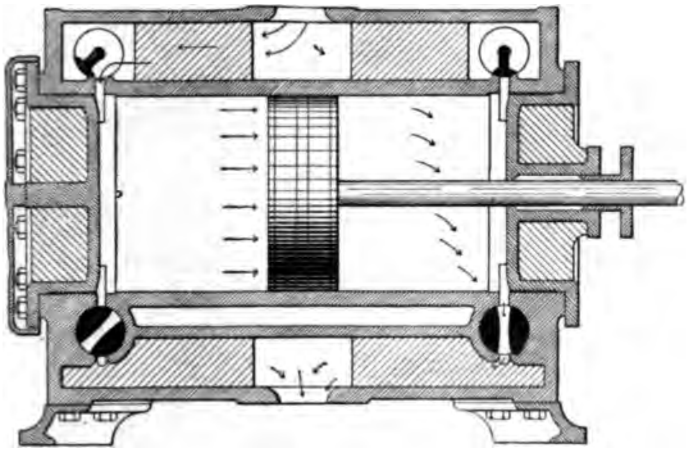


Fig. 24.

The admission valves are generally placed at the top and the exhaust valves at the bottom of the cylinder. Any water which may be formed by steam condensing can readily drain off by this arrangement. There are a great many modifications of the Corliss gear. Fig. 25 is a Harris Corliss Engine.

The advantage of the Corliss gear is the great range through

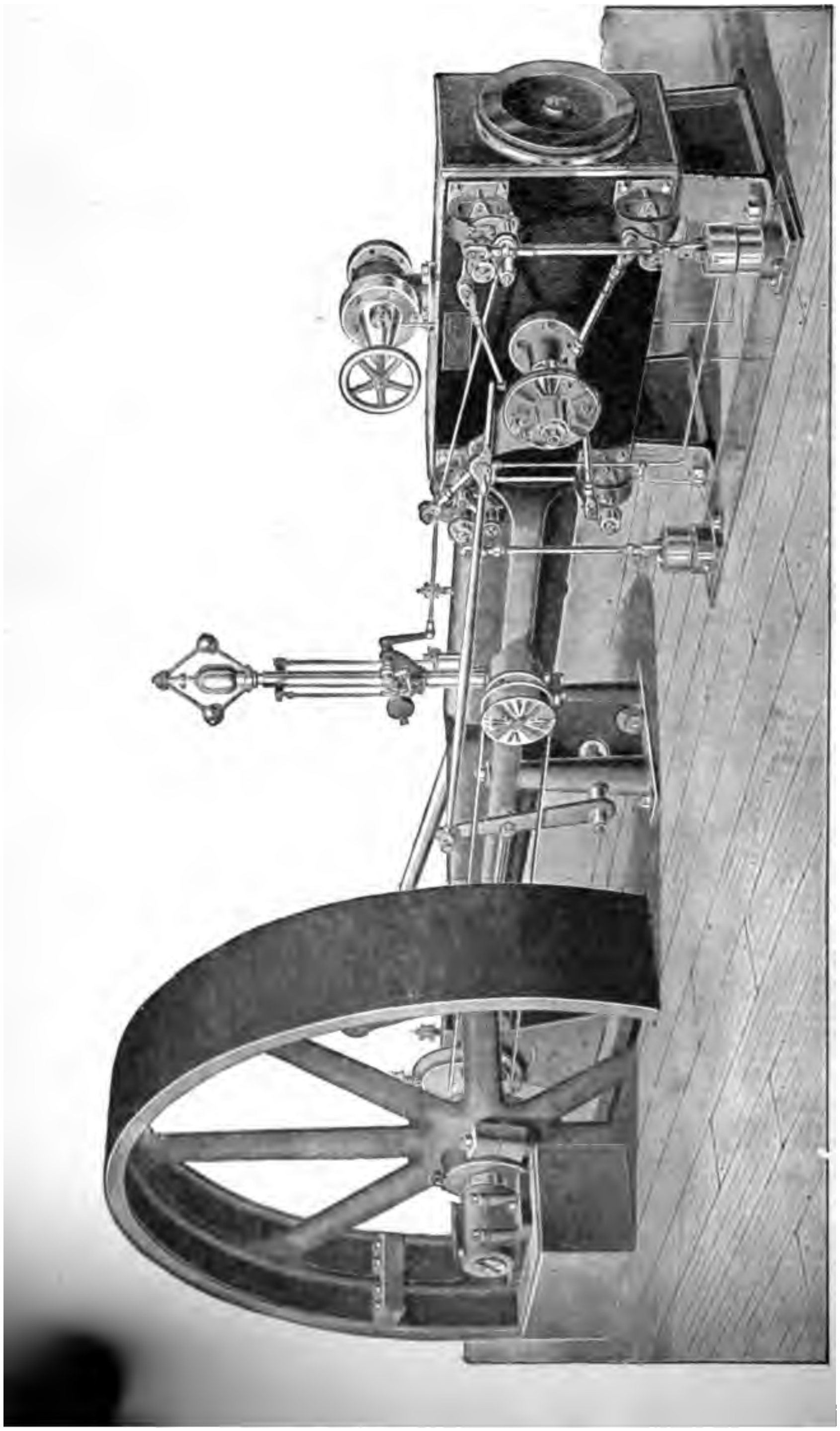


Fig. 25.

which the cut-off can be varied, from very early to very late in the stroke. Another great advantage is the quick action which reduces wire-drawing. To understand fully the loss caused by wire-drawing requires a knowledge of higher mathematics.

CONDENSERS.

When low-pressure steam is cooled it gives up its latent heat; that is, it changes from a vapor to a liquid. We know that a liquid occupies much less space than an equal weight of vapor; hence, by changing the steam to water the pressure is greatly reduced. By cooling the steam in the cylinder in front of the piston the back pressure, or resistance, is decreased. This reduces the pressure necessary to push the piston through the stroke, therefore less steam is required to do the work. This cooling is accomplished by some form of condenser.

There are two types of condensers, surface and jet. A surface condenser is one in which the steam passes through pipes surrounded by water or the water flows through pipes surrounded by steam. In the jet condenser the steam is condensed by coming in contact with cold water, which enters as spray. In both types the steam is condensed to water and a partial vacuum is formed, because water occupies much less space than does an equal weight of steam. If it were not for the air in the entering steam there would be an almost perfect vacuum. For this reason every condenser is fitted with an air pump to remove this air and the condensed steam.

Surface Condensers. The condenser shown in section in Fig. 26 is a common form of the surface type, in which the air pump and circulating pump are both direct acting and are operated by the same steam cylinder. The cold condensing water is drawn from the supply into the circulating or water pump. This pump forces the water up through the valves and water inlet to the condenser. It flows, as indicated by the arrows, through the inner tubes of the lower section, then back through the space between the inner and outer tubes. The water then passes upward and through the upper section, as it did in the lower. It then passes out of the condenser through the water outlet, taking with it the heat it has received from the steam.

The exhaust steam from the engine enters at the exhaust inlet and comes in contact with the perforated plate, which scatters it among the tubes. This method protects the upper tubing from the effect of direct contact with the exhaust steam. The steam expanding in the condenser comes in contact with the cool tubes, through which cold water is circulating, and condenses. The air pump draws the air and condensed steam out of the condenser and thus maintains a partial vacuum. This causes the exhaust steam in the engine cylinder to be drawn into the condenser.

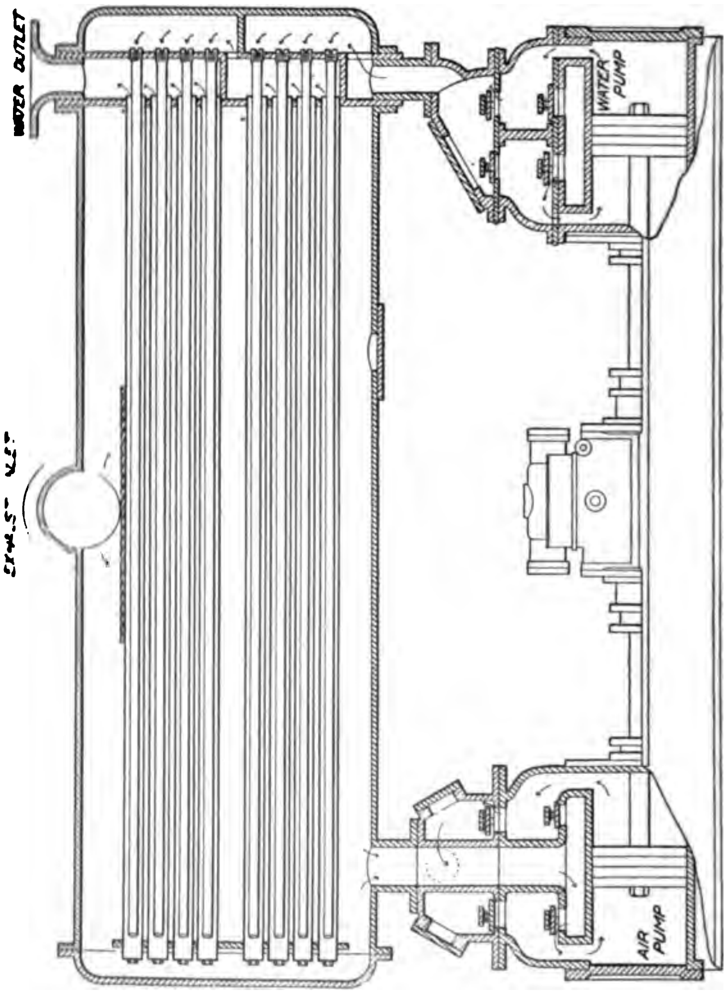


Fig. 26.

The condensed exhaust steam collects at the bottom of the condenser, is drawn into the air pump cylinder and is discharged while heated to the hot well of the boiler. The use of this hot water as feed water is a considerable saving; but the great advantage of the condenser is the reduction in back pressure.

Hot water cannot be used by an ordinary pump as well as cold water because of the pressure of the vapor which arises from the hot water. In the condenser shown, the water and air pumps are run by the piston in the steam cylinder. Sometimes these pumps are connected to the main engine and receive motion from the shaft or crosshead.

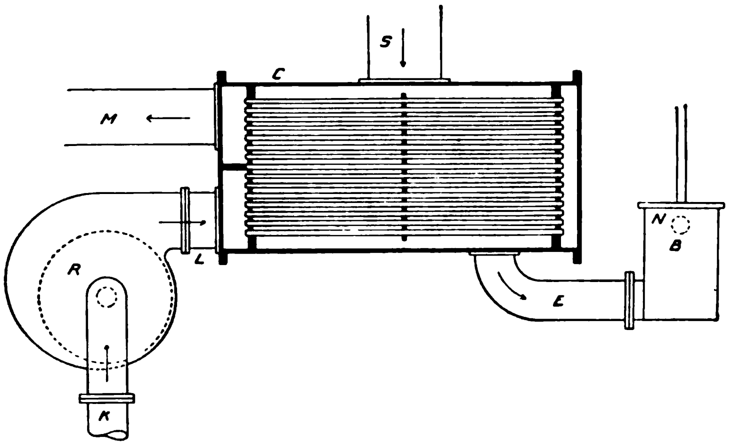


Fig. 27.

The general arrangement of the surface condenser with the necessary pumps is shown in Fig. 27. The cooling water enters through the pipe K, and flows to the circulating pump R, which forces the water into the condenser through the pipe L. In case the water enters the condenser under pressure from city mains no circulating pump is necessary. After flowing through the tubes it leaves the condenser by means of the exit M, and flows away. Exhaust steam enters at S, and is condensed by coming in contact with the cold tubes; the water (condensed steam) then falls to the bottom of the condenser and flows to the air pump B by the pipe E. The air pump removes the air, vapor and condensed steam from the condenser and forces it through the pipe N into the hot well, from which it goes to the boilers or to the feed tank.

The circulating pump, when separate from the condenser, is usually of the centrifugal type. This pump consists of a fan or wheel which is made up of a central web or hub, and arms or vanes. This pump is shown in Fig. 28. The vanes are curved, and as the water is drawn in at the central part the vanes throw it off at the circumference. A suitable casing directs the flow. This type of pump is advantageous because there are no valves to get out of order, and as the lift is little, if any, the pump will discharge a large volume of water in a nearly constant stream. The circulating pump is usually so placed that the water flows to it under a slight head. The pump is driven by an independent engine so that the circulating water may cool the condenser even if the main engine is not working.

The centrifugal pump works more smoothly and with less trouble than an ordinary force pump, because it is not reciprocating and it has no valves.

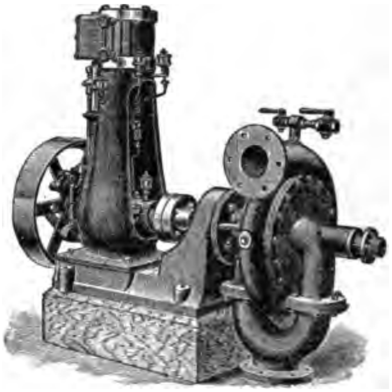


Fig. 28.

Jet Condensers. Fig. 29 shows the longitudinal section of an independent jet condenser with the pump. The cold water used to condense the steam enters at A, passes down the spray pipe B, and is broken into a fine spray by means of the spray cone C. This action insures a rapid and thorough mixing of the steam and water and consequently a rapid condensation. The exhaust steam enters at D, with a comparatively high velocity, which is imparted to the water. The whole mixture of water, steam and vapor passes at high velocity through the conical chamber E to the pump cylinder F. The pump forces this water to the pipe G. The spray cone is adjusted by the stem which passes through the stuffing box at the top of the condenser. The valves are shown at H and K. The steam end of the pump is at L. Motion of the valve is produced by the rocker arm J.

In Fig. 30 a jet condenser is shown connected to a stationary engine and boiler. The exhaust pipe leads from the engine to the condenser, the arrows indicating the direction of flow. Cold water enters the condenser through a pipe connected to the well. Part of the mixture of exhaust steam and condensed water goes to the feed-water heater, which is kept nearly full; the rest passes to the sewer. The heater is placed a little above the feed pump, so that the water will enter the pump under a slight head, because the pump cannot raise water warmed by exhaust steam as readily as cold water.

The surface condenser is used almost universally in marine practice. Its first cost is more than that of the jet condenser and it requires more condensing water, but it allows only the condensed steam to return to the boiler. It is also used in stationary work when the condensing water is very impure. The jet condenser is not adapted for marine work, as it pumps both the condensing water and the condensed steam to the hot well. Hence, if salt water or water containing lime is used, it will enter the boiler and form sediment and scale. This type is used where fresh and moderately pure water is available.

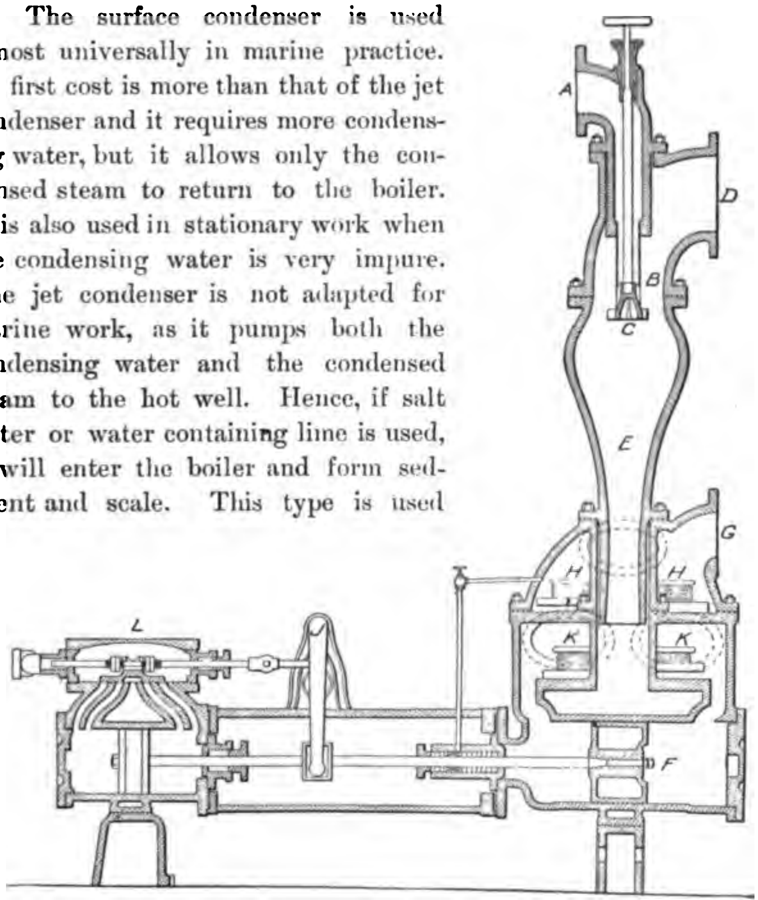


Fig. 29.

It has been mentioned before that Watt always condensed the exhaust steam from his engines, and that when higher pres-

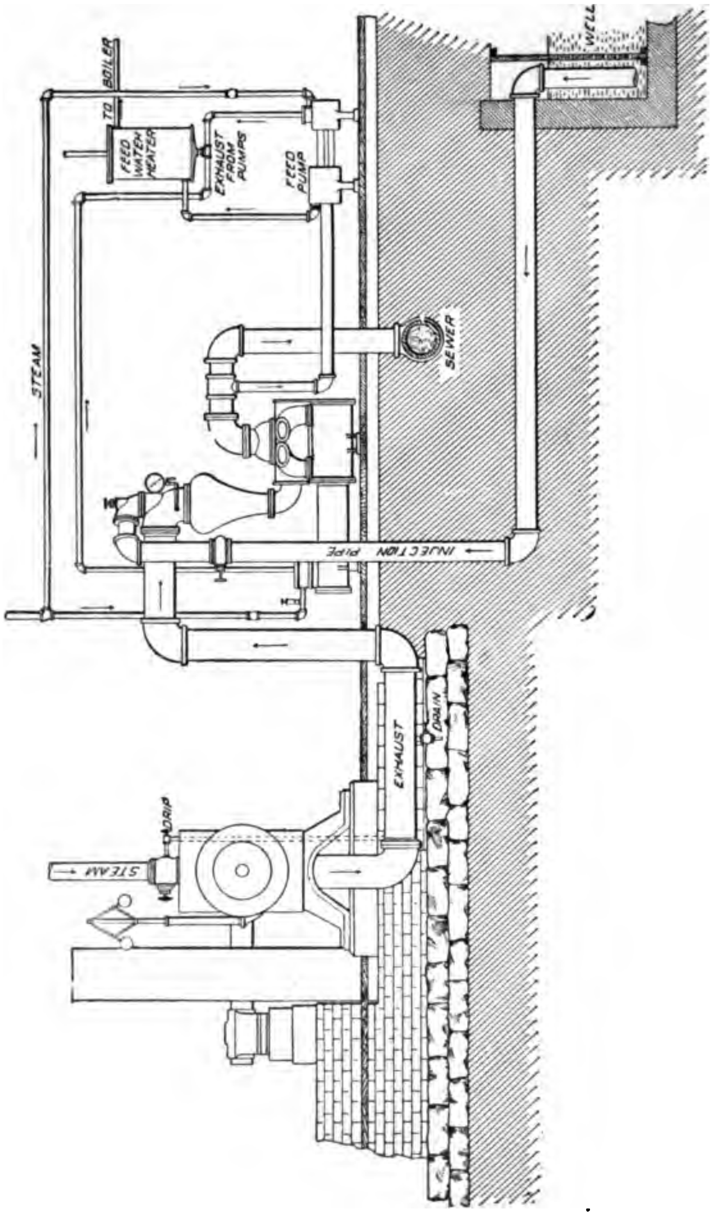

TO BOILER
FEED WATER HEATER
EXHAUST FROM PUMPS
FEED PUMP
STEAM
SEWER
INJECTION PIPE
WELL
EXHAUST
DRAIN
DRIP
STEAM

Fig. 30.

sures came into use some makers let the steam discharge into the atmosphere. This leads to the distinction between condensing and noncondensing engines. Both types are in common use, but the condensing engines are much more economical than the noncondensing, as far as fuel is concerned; but to condense the steam, considerable water is necessary, and condensing engines cost more and require more care. Consequently in some cases it is quite as economical, all things considered, to use the noncondensing engine.

COOLING TOWERS.

It sometimes happens that it is impossible to place a steam plant in close proximity to a natural water supply. In such cases the water necessary for the condenser (the circulating water) is expensive, and if the cost is very great it does not pay to add the condenser, because the cost of the circulating water might more than offset the gain from condensing. If, however, some means could be provided whereby the circulating water as it issues from the condenser could be cooled and then used over again in the condenser, the noncondensing engine could then be run condensing; thus taking advantage of all the benefits due to the use of reduced back pressure and heating of the feed water. This has been attempted by conducting the heated discharge water to a pond, where it is allowed to cool to a lower temperature before being used again. Another plan is to place in the yard or on the roof of the building large shallow pans, in which the water is cooled by being exposed to the atmosphere. These methods are unsatisfactory on account of the considerable area necessary and the slow action. In addition they are uncertain, because they are dependent upon atmospheric conditions.

A more efficient and at the same time more expensive process is to use a cooling tower or a water table. Fig. 31 shows the general arrangement of a cooling tower located upon the roof of a building. The discharge from the condenser is led, as shown by the arrows, to the top of the cooling tower, where it is cooled before being returned to the condenser. This cooling is effected by distributing the water, by a system of piping, to the upper edge of a series of mats or slats, over the surface of which the water

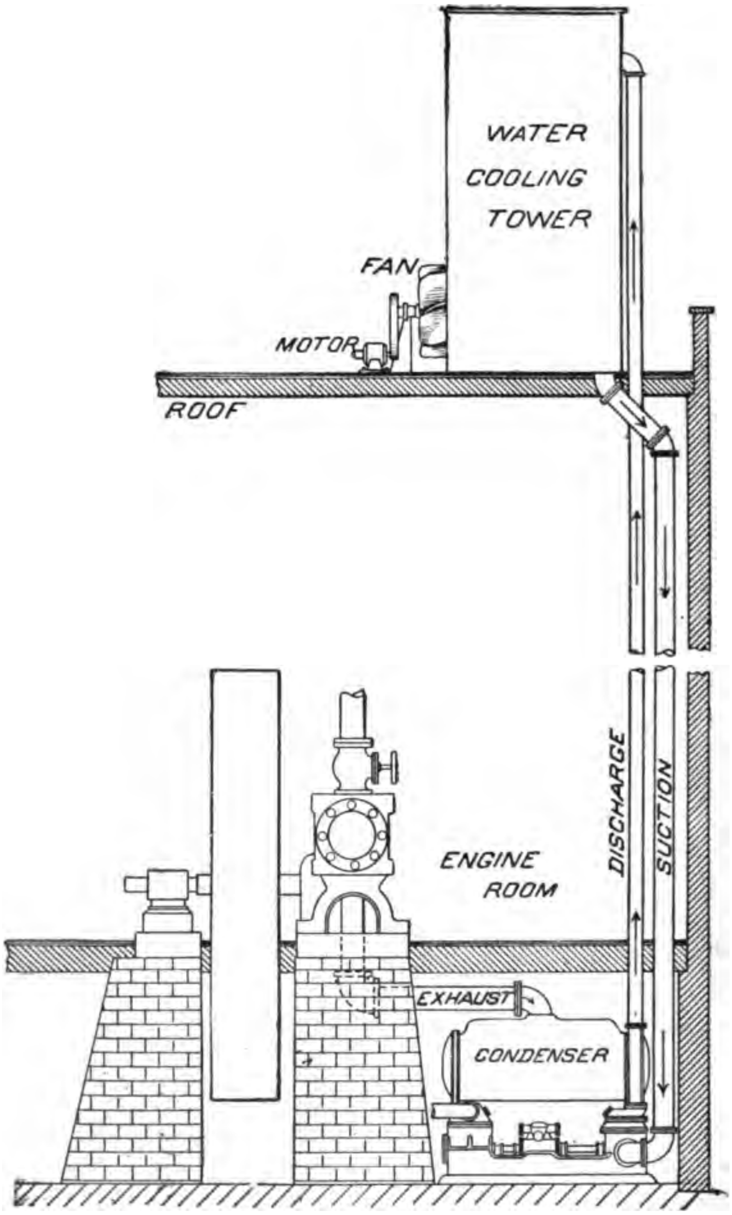

WATER
COOLING
TOWER
FAN
MOTOR
ROOF
ENGINE
ROOM
DISCHARGE
SUCTION
EXHAUST
CONDENSER

Fig. 31.

flows in a thin film to a reservoir which is situated in the bottom of the cooling tower. The mats partially interrupt the flow, and by breaking up the water in small streams cause new portions to be exposed to the cooling effect of the air currents. The water from the reservoir then flows downward through the suction pipe, and is pumped by the circulating pump through the condenser. After passing through the condenser and absorbing heat from the exhaust steam, it rises through the discharge pipe and commences the circuit over again.

The tower may have several arrangements and be made of various materials. A satisfactory form is constructed of steel plates; within the tower are a large number of mats of steel wire cloth galvanized after weaving.

To assist in the cooling of the water, the air is often made to circulate rapidly by means of a fan, which forces the air into the lower part of the tower and upwards among the mats. This fan is usually of the ordinary type, and may be driven by an electric motor, a line of shafting, or by a small independent engine.

In case the fan is not used, the mats are arranged so that they are exposed to the atmosphere, as shown in Fig. 32. This of course necessitates the removal of the steel casing. Usually the fanless tower must be placed at the top of a high building, or in some position where the currents of air can readily circulate among the mats.

With an efficient type of cooling tower the water may be reduced from 30 to 50°, thus allowing a vacuum of from 22 to 26 inches. This will of course greatly increase the economy of the plant, and allow the heated feed water to be returned to the boiler

The water table is usually made of wooden slats placed in the ground near the plant. After trickling over the slats and becoming cooled by the air, it collects in the bottom of the reservoir and is then pumped into the condenser.

THE FLY WHEEL.

It is evident that while the piston can push the crank around during part of the stroke, and pull it during another part, there are still two places (called dead points) at the ends of the stroke, where the pressure on the piston, no matter how great, can exert

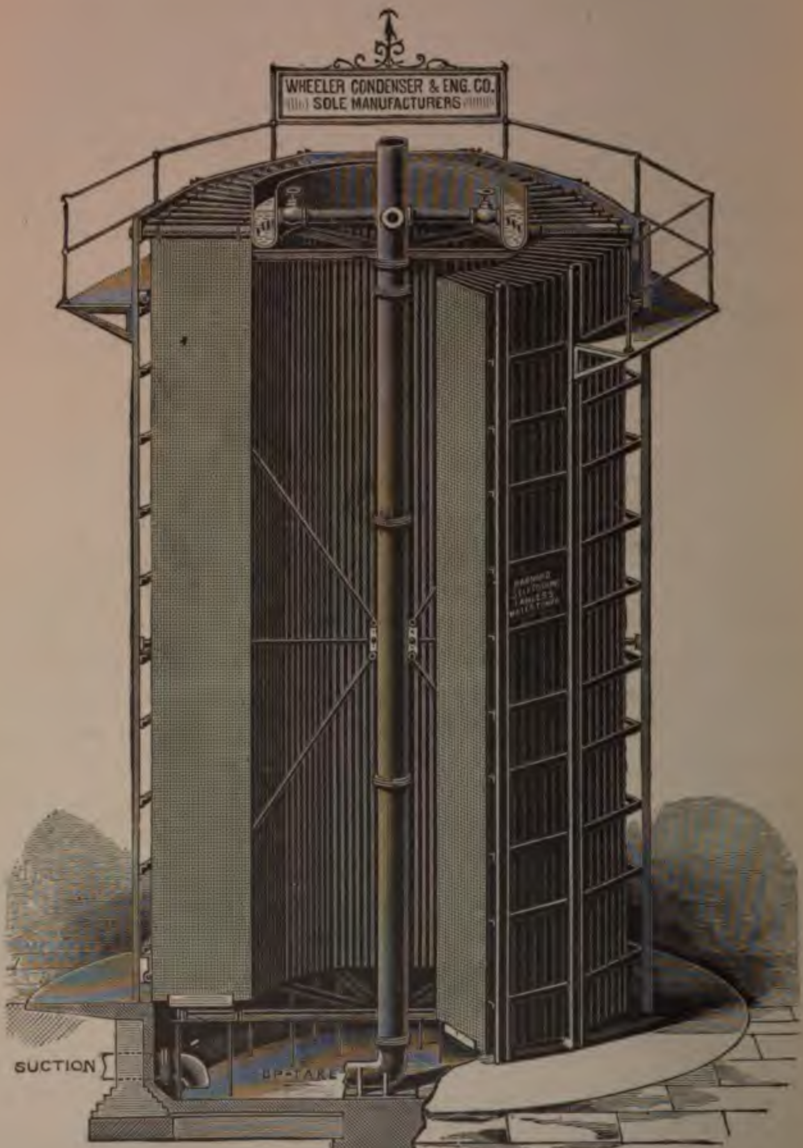

WHEELER CONDENSER & ENG. CO.
SOLE MANUFACTURERS
SUCTION
UPTAKE

Fig. 32

no turning moment on the shaft. Therefore, if some means is not provided for making the shaft turn past these points without the assistance of the piston, it may stop. This means is provided in the fly wheel, which is merely a heavy wheel placed on the shaft. On account of the momentum of the fly wheel it cannot be stopped quickly, and therefore carries the shaft around until the piston can again either push or pull.

If a long period be considered, the mean effort and the mean resistance must be equal; but during this period there are temporary changes of effort, the excesses causing increase of speed. To moderate these fluctuations several methods are employed.

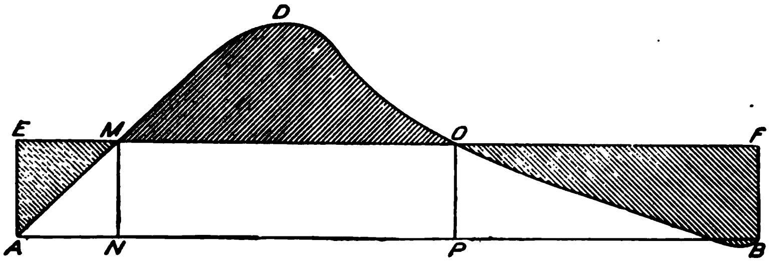


Fig. 33.

The turning moment on the shaft of a single cylinder engine varies, first, because of the change in steam pressure, and second, on account of the angularity of the connecting rod. Before the piston reaches mid-stroke the turning moment is a maximum, as shown by the curve, (Fig. 33). While near the ends of the stroke (the dead points) the turning moment diminishes and finally becomes zero. This, of course, tends to cause a corresponding change in the speed of rotation of the shaft. In order to have this speed as nearly constant as possible, and to give a greater uniformity of driving power, the engine may be run at high speed. By this means the inertia of the revolving parts, such as the connecting rod and crank, causes less variation. When the work to be done is steady and always in the same direction (as in most factories), a heavy fly wheel may be used.

The heavier the fly wheel, the steadier will be the motion. It is, of course, desirable in all factory engines to have steady motion, but in some it is more important than in others. For instance, in a cotton mill it is absolutely necessary that the machinery shall move with almost perfect steadiness; consequently mill engines always have very large, heavy fly wheels. It is undesirable to use larger wheels than are absolutely necessary, because of the cost of the metal, the weight on the bearings and the danger from bursting.

If the turning moment which is exerted on the shaft from the piston could be made more regular, and if dead points could be avoided, it would be possible to get a steadier motion with a much smaller fly wheel.

If the engine must be stopped and reversed frequently, it is better to use two or more cylinders connected to the same shaft. The cranks are placed at such angles that when one is exerting its minimum rotative effort, the other is exerting its maximum turning moment; or, when one is at a dead center, the other is exerting its greatest power. These two cylinders may be identically the same, as is the case with most hoisting engines and with many locomotives; or the engine may be compound or triple expansion.

This arrangement is also used on engines, for mines, collieries, and for hoisting of any sort where ease of stopping, starting and reversing are necessary. Simple expansion engines with their cranks at right angles are said to be coupled.

The governor adjusts the power of the engine to any large variation of the resistance. The fly wheel has a duty to perform which is similar to that of the governor. It is designed to adjust the effort of the engine to sudden changes of the load which may occur during a single stroke. It also equalizes the variation in rotative effort on the crank pin. The fly wheel absorbs energy while the turning moment is in excess of the resistance and restores it while the crank is at or near the dead points. During these periods the resistance is in excess of the power.

The action of the fly wheel may be represented as in Figs. 33 and 34. It will be noticed that in Fig. 33 the curve of crank effort runs below the axis toward the end of the stroke. This is because the compression is greater than the pressure near the end of expansion, and produces a resultant pressure on the piston. In Fig. 34 the effect of compression has been neglected. Let us suppose that the resistance, or load, is uniform. In Fig. 33 the line

A B is the length of the semi-circumference of the crank pin, or it is the distance the crank pin moves during one stroke. The curve A M D O B is the curve of turning moment for one stroke. M N is the mean ordinate, and therefore A E F B represents the constant resistance. The effort and resistance must be equal if the speed is uniform; hence A E F B = A M D O B. Then area A E M + area O F B = area M D O. At A the rotative effort is zero because the crank pin is at the dead point; from A to N the turning moment is less than the resistance. At N the resistance and the effort are equal. From N to P the effort is in excess of the resistance. At P the effort and resistance are again equal. From P to B the resistance is greater than the effort. In other words, from A to N the work done by the steam is less than the resistance. This shows that the work represented by the area A E M must have been done by the moving parts of the engine. From N to P the work done by the steam is greater than the resistance, and the excess of energy is absorbed by or stored in the moving parts. From P to the end of the stroke, the work represented by the area O F B is done on the crank pin by the moving parts.

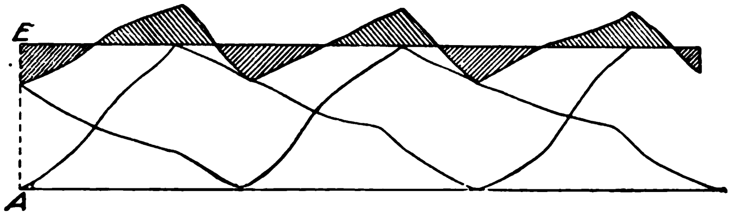


Fig. 34.

We know from the formula, $E = \frac{W V^2}{2g}$, that energy is proportional to the square of the velocity. Hence as W and g remain the same, the velocity must be reduced when the moving parts are giving out energy, and increased when receiving energy. Thus we see that the tendency of the crank pin is to move slowly, then more rapidly. The revolving parts of an engine have not sufficient weight to store this surplus energy, hence a heavy fly wheel is used.

In case there are two engines at right angles, two effort curves must be drawn, as shown in Fig. 34. The mean ordinate A E is equal to the mean or constant resistance. There are two minimum and two maximum velocities in one stroke. The diagram shows that the variation is much less than for a single cylinder; hence a lighter fly wheel may be used.

The *weight of the fly wheel* depends upon the character of the work done by the engine. For pumping engines and ordinary machine work the effort need not be as constant as for electric lighting and fine work. In determining the weight of a fly wheel the diameter of the wheel must be known, or the ratio of the diameter of the wheel to the length of stroke. If the wheel is too large, the high linear velocity of the rim will cause too great a centrifugal force and the wheel is likely to break. In practice, about 6,000 feet per minute is taken as the maximum linear velocity of cast-iron wheels. When made of wood and carefully put together, the velocity may be taken as 7,000 to 7,500 feet per minute.

We know that linear velocity is expressed in feet per minute by the formula, $V = 2\pi R N$, or $V = \pi D N$.

Then if a wheel runs at 100 revolutions per minute, the allowable diameter would be,

$$6{,}000 = 3.1416 \times D \times 100$$

$$D = \frac{6{,}000}{3.1416 \times 100} = 19.1 \text{ feet.}$$

If a wheel is 12 feet in diameter the allowable speed is found to be,

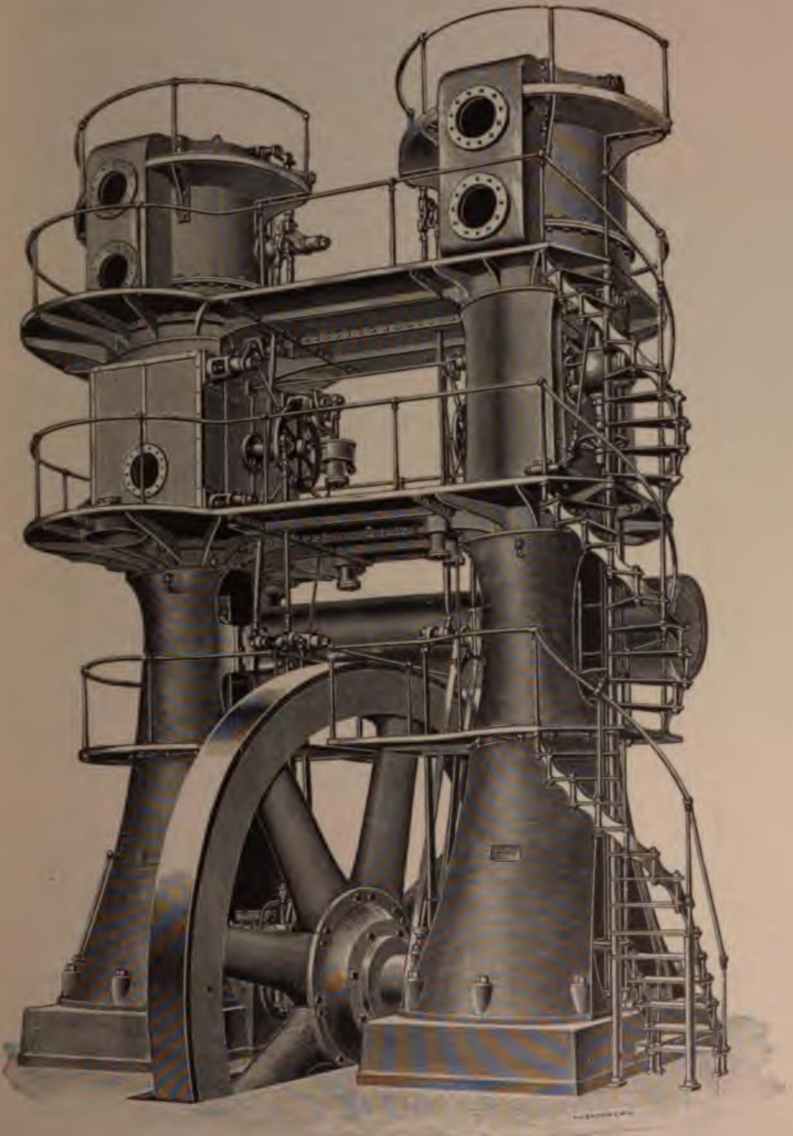
$$N = \frac{V}{\pi D}$$

$$= \frac{6{,}000}{3.1416 \times 12} = 159 \text{ revolutions per minute.}$$

It is usual to make the diameter a little less than the calculated diameter.

Having determined the diameter, the weight may be calculated by several methods. There are many formulas to obtain this result given by various authorities. One formula is given as follows:

$$W = \frac{C \times d^2 \times b}{D^2 \times N^2}$$



VERTICAL COMPOUND BLOWING ENGINE
ALLIS-CHALMERS COMPANY

THE NEW YORK
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ASTOR, LENOX
TILDEN FOUNDATIONS

In the above, $W =$ weight of the rim in pounds
$d =$ diameter of the cylinder in inches
$b =$ length of stroke in inches
$D =$ diameter of fly wheel in feet
$N =$ number of revolutions per minute

C is a constant which varies for different types and conditions.

Slide-valve engines, ordinary work, $C = 350,000$
Corliss engines, ordinary work, $C = 700,000$
Slide-valve engines, electric lighting, $C = 700,000$
Corliss engines, electric lighting, $C = 1,000,000$
Automatic high-speed engines, $C = 1,000,000$

Example. Let us find the weight of a fly-wheel rim for an automatic high-speed engine used for electric lighting. The cylinder is 24 inches in diameter; the stroke is 2 feet. It runs at 300 revolutions per minute, and the fly wheel is to be 6 feet in diameter.

$$W = 1,000,000 \times \frac{(24)^2 \times 24}{36 \times 90,000}$$

$$W = 4,266 \text{ pounds}$$

Another example. A plain slide-valve engine for electric lighting is 20″ × 24″. It runs at 150 revolutions per minute. The fly wheel is to be 8 feet in diameter. What is the weight of its rim?

$$W = 700,000 \times \frac{400 \times 24}{64 \times 22,500}$$

$$W = 4,666 \text{ pounds.}$$

The weight of a fly wheel is considered as being in the rim. The weight of the hub and arms is simply extra weight. Then, if we know the weight of the rim and its diameter, we can find the width of face and thickness of rim. Let us assume the given diameter to be the mean of the diameter of the inside and outside of the rim.

Let $b =$ width of face in inches
$t =$ thickness of rim in inches
$d =$ diameter of fly wheel in inches
$.2607 =$ weight of 1 cubic inch of cast iron

Then,

$$W = .2607 \times b \times t \times \pi d$$
$$= b \times t \times .819\, d$$

Suppose the rim of a fly wheel weighs 6,000 pounds, is 9 feet in diameter, and the width of the face is 24 inches. What is the thickness of the rim?

$$t = \frac{W}{.819\ d\ b}$$
$$= \frac{6{,}000}{.819 \times 108 \times 24}$$
$$= 2.83 \text{ inches}$$

In this case the rim would probably be made $2\frac{13}{16}$ inches thick. The total weight, including hub and arms, would probably be about 8,000 pounds.

GOVERNORS.

The load on an engine is never constant, although there are cases where it is nearly uniform. While the engine is running at constant speed, the resistance at the fly-wheel rim is equal to the work done by the steam. If the load on the engine is wholly or partially removed, and the supply of steam continues undiminished, the force exerted by the steam will be in excess of the resistance. Work is equal to force multiplied by distance; hence, with constant effort, if the resistance is diminished, the distance must be increased. In other words, the speed of the engine will be increased. The engine will "race," as it is called. Also, if the load increases and the steam supply remains constant, the engine will "slow down."

It is evident, then, that if the speed is to be kept constant some means must be provided so that the steam supply shall at all times be exactly proportional to the load. This is accomplished by means of a governor.

Steam-engine governors act in one of two ways: they may regulate the *pressure* of steam admitted to the steam chest, or they may adjust the speed by altering the *amount* of steam admitted. Those which act in the first way are called throttling governors, because they throttle the steam in the main steam pipe. Those of the latter class are called automatic cut-off governors, since they automatically regulate the *cut-off*.

Theoretically, the method of governing by throttling the steam causes a loss in efficiency, but the throttling superheats the steam, thus reducing cylinder condensation. By the second method the loss in efficiency is very slight, unless the ratio of expansion is already great, in which case shortening the cut-off causes more cylinder condensation. This subject will be taken up in detail later.

In most governors, centrifugal force, counteracted by some other force, is employed. A pair of heavy masses (usually iron balls or weights) are made to revolve about a spindle, which is driven by the engine. When the speed increases, centrifugal force increases, and the balls tend to fly outward; that is, they revolve in a larger circle. The controlling force, which is usually gravity or springs, is no longer able to keep the balls in their former path. When, therefore, the increase is sufficiently great, the balls in moving outward act on the regulator, which may throttle the steam or cause cut-off to occur earlier.

With the throttling governor a balanced throttle valve is placed in the main steam pipe leading to the valve chest. If the engine runs faster than the desired speed, the balls are forced to revolve at a higher speed. The increase in centrifugal force will cause them to revolve in a larger circle and in a *higher plane.* By means of some mechanism (levers and gears) the spindle may be forced downward, thus partially closing the valve. The engine, therefore, takes steam at a lower pressure, and the power supplied being less, the speed falls slightly.

Similarly, if the load is increased, the engine slows down, which causes the balls to drop and open the valve more widely, steam at higher pressure is then admitted, and the speed is increased to the regular number of revolutions.

With the Corliss or Wheelock engine the governor of this type acts differently. Instead of throttling the steam in the steam pipe, the governor is connected to the releasing gear by rods. An increase of speed causes the releasing gear to unhook the disengaging link earlier in the stroke. This causes earlier cut-off, which of course decreases the power and the speed, since the amount of steam admitted is less. If for any reason the load increases, the governor causes the valves to be held open longer. The cut-off, therefore, occurs later in the stroke.

One of the most common forms of governor is similar to that invented by James Watt. It is called, from its appearance, the pendulum governor. It is shown in Fig. 35. To consider the theory of the pendulum governor, the masses of the balls are assumed to be concentrated at their centers, and the rods made of some material having no weight.

When the governor is revolving about its axis at a constant speed the balls revolve in a circle having a radius r The distance from this plane to the intersection of the rods, or the rods produced, is called the height and is equal to h.

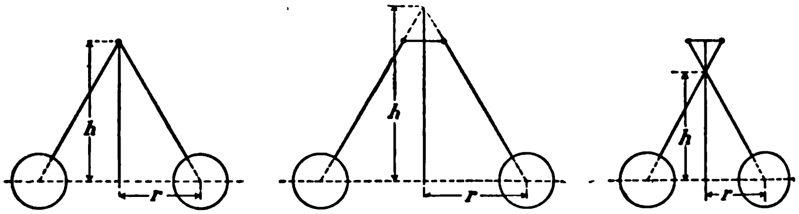

Fig. 35.

If the balls revolve faster, the centrifugal force increases, r becomes greater and h diminishes. We know that centrifugal force is expressed by the formula,

$$F = \frac{Wv^2}{g\,r}$$

Then centrifugal force varies inversely as the radius.

While the pendulum is revolving, centrifugal force acts horizontally outward and tends to make the balls fly from the center; gravity tends to make the balls drop downward. In order that the balls shall revolve at a certain height, the moments of these two forces about the center must be equal, or the weight of the balls multiplied by their distance from the center must equal the centrifugal force multiplied by the height, or.

$$W \times r = F \times h$$

from which, $$\frac{h}{r} = \frac{W}{F}$$

or, $$\frac{h}{r} = \frac{W}{\frac{Wv^2}{g\,r}} = \frac{g\,r}{v^2}$$

from which, $$h = \frac{g\,r^2}{v^2}$$

Now we know that the linear velocity of a point revolving in the circumference of a circle is expressed as $2 \pi r N$ feet per second.

Then, $$h = \frac{g r^2}{v^2} = \frac{g r^2}{4\pi^2 r^2 N^2} = \frac{g}{4\pi^2 N^2}$$

Since we know the values of g and π we can write the formula,

$$h = \frac{32.16}{4 \times 3.1416^2 \times N^2} = \frac{.8146}{N^2} \text{ feet per second, or}$$

$$h = \frac{9.775}{N^2} \text{ inches per second.}$$

If N is the number of revolutions per minute, since $60^2 = 3600$,

$$h = \frac{2{,}932.56}{N^2} \text{ feet}$$

$$h = \frac{35{,}190.7}{N^2} \text{ inches}$$

From the above formula, we see that the height is independent of the weight of the balls or the length of the rod; it depends upon the number of revolutions. The height varies inversely as the square of the number of revolutions.

The ordinary pendulum governor is not isochronous; that is, it does not revolve at a uniform speed in all positions; the speed changes as the angle between the arms and the spindle changes.

The early form consisted of two heavy balls suspended by links from a pin connection in a vertical spindle, as shown in Figs. 36 and 37. The spindle is caused to revolve by belting or gearing from the main shaft, so that as the speed increases, centrifugal force causes the balls to revolve in a circle of larger and larger diameter. The change of position of these balls can be made to affect the controlling valves so that the admission or the throttling shall vary with their position. With this governor it is evident that for a given speed of the engine there is but one position possible for the governor; consequently one amount of throttling or one point of cut-off, as the case may be. If the load varies, the speed of the engine will change. This causes the position of the governor balls to be changed slightly, thus altering the pressure

But in order that the pressure or cut-off shall remain changed, the governor balls must stay in their new position. That is to say, the speed of the engine must be slightly changed. Thus with the old ball governors there was a slightly different speed for each load. This condition has been greatly improved by various modifications until now such governors give excellent regulation.

While the engine is running with a light load, the valve controlled by the governor will be open just enough to admit steam at a pressure that will keep the engine running at a given speed. Now, if the engine is heavily loaded, the throttle valve must be wide open. The change of opening is obtained by a variation in the height of the governor, which is caused by a change of speed. Thus we see that the governor can control the speed only within certain limits which are not far apart. The difference in the extreme heights of the governor must be sufficient to open the throttle its entire range. In most well-designed engines the speed will not vary more than 4 per cent; that is, 2 per cent above or below the mean speed.

From the formula $h = \frac{35{,}190.7}{N^2}$, we can compute the heights corresponding to given speeds as shown by the following table:

Number of Revolutions per Minute.	Height in Inches.	Variation of Height in Inches 4 per cent.
250	.563	.0225
200	.879	.035
175	1.149	.046
150	1.564	.062
125	2.252	.090
100	3.519	.140
75	6.256	.250
50	14.076	.563

In the above table the second column is found from the formula $h = \frac{35{,}190.7}{N^2}$. The third column is the variation in height for a speed variation of 4 per cent or 2 per cent either above or below the mean.

From the table we see that for a considerable variation of speed there is but slight variation in the height of the governor. Also for high speeds the height of the governor is so small that it would be difficult to construct it. The slight variation in height is too small to control the cut-off or throttling mechanism throughout the entire range.

Other disadvantages of the fly-ball governor are as follows: it is apparent that the valves must be controlled by the weight of the governor balls. In large engines this requires very heavy balls in order to quickly overcome the resistance of the valves. But these large balls have considerable inertia and will therefore be reluctant to change their speed with that of the engine. The increased weight will also increase the friction in the governor joints, and the cramping action existing when the balls are driven by the spindle will increase this friction still further. All these things tend to delay the action of the governor, so that in all large engines the old-fashioned governor became sluggish. The balls had to turn slowly because they were so heavy; this was especially troublesome in high-speed engines.

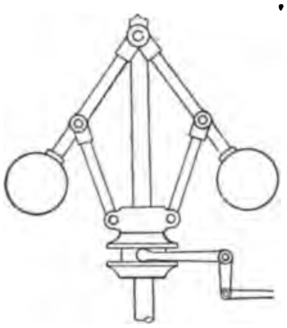
Fig. 36.

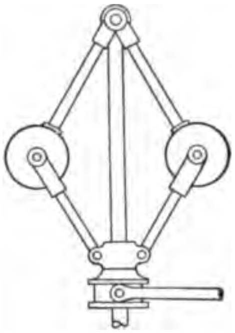
Fig. 37.

To remedy these defects the weighted or Porter governor was designed. (See Fig. 38.) It has a greater height for a given speed, and the variation in height for a given variation of speed is greater. When a governor has this latter quality, that is, a great variation in height for a given variation of speed, it is said to be sensitive. By increasing this variation in height the sensitiveness is increased. Thus, if a governor running at 50 revolutions has a variation in height of .57 inch, it is not as sensitive as one having a variation of 1 inch for the same speed.

In the weighted governor, the weight is formed so that the center of gravity is in the axis. It is placed on the spindle and is free to revolve. The weight adds to the weight of the balls, and thus increases the moment of the weight. It does not, however, add to the centrifugal force, and hence the moment of this force is unchanged. We may then say the weight adds effect to the weight but not to the centrifugal force, and as a consequence the height of the governor, for a given speed, is increased. If we let W equal the combined weight of the balls as before, and W′ equal the added weight, the moments are,

$$(W + W') \times r = F \cdot h$$

$$(W + W') \times r = \frac{W v^2}{g r} \times h$$

$$h = (W + W') r \times \frac{g r}{W v^2}$$

$$= \frac{(W + W') r^2 g}{W \times 4 \pi^2 r^2 N^2}$$

$$= \frac{(W \times W')}{W} \times \frac{g}{4 \pi^2 N^2}$$

We know that $\frac{g}{4 \pi^2 N^2} = \frac{.8146}{N^2}$.

Then $h = \left\{ \frac{W + W'}{W} \right\} \times \frac{.8146}{N^2}$.

Hence the height of a weighted governor is equal to the height of a simple pendulum governor multiplied by

$$\left\{ \frac{W + W'}{W} \right\} \text{ or } \left\{ 1 + \frac{W'}{W} \right\}.$$

For instance, if the height of a simple pendulum is 10 inches, and the weight of the balls equal to the added weight, the height will be,

$$h = \left\{ \frac{1 + 1}{1} \right\} \times 10$$

$$= 2 \times 10$$

Thus we see that if a weight equal to the combined weight of the balls is added, the height of the governor will be doubled.

We know that if the balls fall, the cut-off comes later. If the belt driving the governor slips off or breaks, the balls will drop, and, making the cut-off later, will allow the engine to "run away." To diminish this danger many governors are provided with some kind of safety stop, which closes the valve when the governor loses its normal action. Usually a trip is provided which the governor does not touch in its normal positions, but which will be released if the balls drop down below a certain point.

In another arrangement, instead of a weight, a strong spring is used, and this makes it possible to put the governor in any position.

Spring Governors. In many cases a spring is used in place of the weight. This type of governor is used frequently on throttling engines; it consists of a pendulum governor with springs added to counteract the centrifugal force of the balls. Thus the height and sensitiveness are increased. Fig. 39 shows the exterior view of a Waters governor, and Fig. 40 the same governor having the safety stop. In this governor the weights are always in the same plane, the variation in height being due to the action of the bell crank levers connecting the balls and spindle. When the balls move outward the spindle moves downward and tends to close the valve. The governor balls revolve by means of a belt and bevel gears. The valve and seat are shown in section in Fig. 41. The valve is a hollow cylinder with three ports, by means of which steam enters the valve. The seat is made in four parts, that is, there are four edges that the steam passes as it enters the valve. The valve being cylindrical and having steam on both sides is balanced, and because of the many openings only a small travel is necessary.

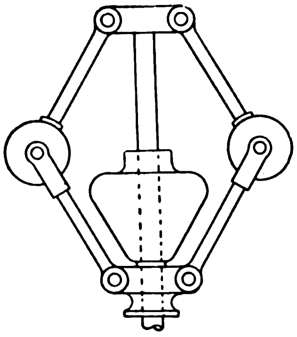

Fig. 38.

Shaft Governors. Usually some form of pendulum governor is used for throttling engines. For governing an engine by varying the point of cut-off, shaft governors are generally used; however, Corliss engines and some others use pendulum governors for this purpose. Cut-off governors are called shaft governors because they are placed on the main shaft; they are made in many forms, but the essential features of all are the same. Two pivoted masses or weights are arranged symmetrically on opposite sides of the shaft, and their tendency to fly outward when the speed increases is resisted by springs. The outward motion of the weights closes the admission valve earlier, and the inward motion closes it later. This change is effected by altering the position of the eccentric, either by changing the eccentricity or the angular advance.

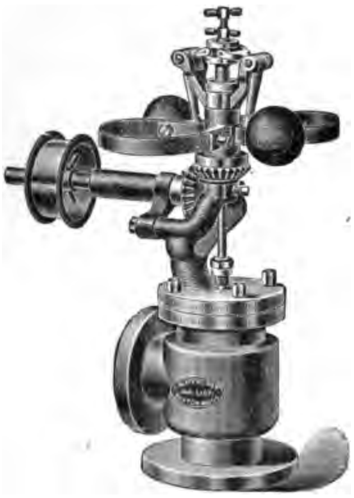

Fig. 39.

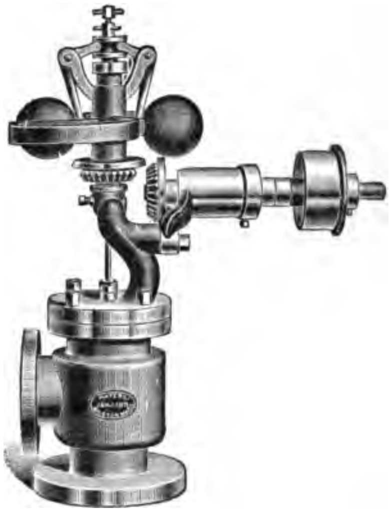

Fig. 40.

Shaft governors are made in a great variety of ways, no two types being exactly alike. If the principles of a few types are understood, it is easy to understand others. The following illustrates two common methods of shifting the eccentric.

Buckeye Engine Governor. The valve of the Buckeye engine is hollow and of the slide valve type. The cut-off valve is inside. The change of cut-off is due to the alteration of the angular advance. The arrangement of the parts which effect the change of angular advance is shown in Fig. 42. A wheel which contains and supports the various parts of the governor is keyed to the shaft. Two arms, having weights A A at the ends, are pivoted to the arms of the wheel at *b b*. The ends having the weights are connected to the collar on the loose eccentric C by means of rods B B.

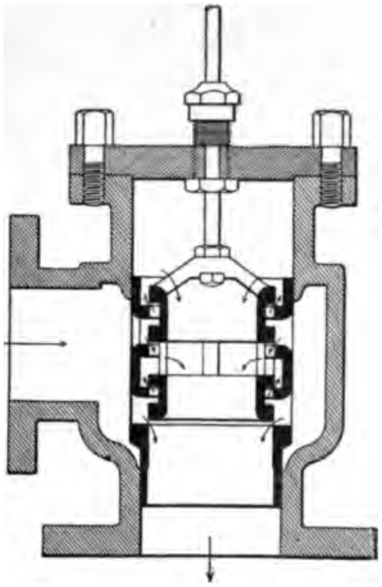

Fig. 41.

When the weights move to the position indicated by the dotted lines, the eccentric is turned on the shaft about a quarter of a revolution in the direction in which the engine runs. That is, the eccentric is advanced or the angular advance is increased. Now we know that if the angular advance is increased, cut-off occurs earlier. This is shown by the table on page 12 of "Valve Gears." If the engine had a single plain slide valve the variation of the angular advance would produce too great a variation of lead; but as this engine has a separate valve for cut-off, admission is not altered by the cut-off valve.

The springs F F balance the centrifugal force of the weights. The weights A A are varied to suit the speed; the tension on the springs is altered by means of the screws *c c*. Auxiliary springs are added in order to obtain the exactness of regulation necessary for electric lighting. These springs tend to throw the arms outward, but act only during the inner half of this movement.

The Straight-Line Engine Governor. Fig. 43 shows the governor of the straight-line engine. It has but one ball, B, which is linked to the spring S and to the plate D E, on which is the eccentric C. When the ball flies outward in the direction indicated by the arrow F, the eccentric is shifted about the pivot O; the links moving in the direction of arrow H. The ball is heavy and at a considerable distance from the center; hence it has a great centrifugal force, and the spring must be stiff.

The governor of the Buckeye engine alters the cut-off by changing the *angular advance*. The straight-line engine governor changes the *travel* of the valve. Shaft governors which alter the cut-off by changing the valve travel are very common.

LUBRICATION.

If two pieces of cast iron, just as they come from the foundry, are rubbed together, they will not slide over each other easily, because of little projections. If this same iron is filed or planed, the pieces will slide much more easily. This is because the rough places have been smoothed, or filled up with dust. If now we put some engine oil on the pieces, they will slide very easily. This is because the more minute depressions have been filled up and the whole surface is made comparatively smooth. No matter how carefully we might plane and polish any surface, a microscope would show that it was still a little rough.

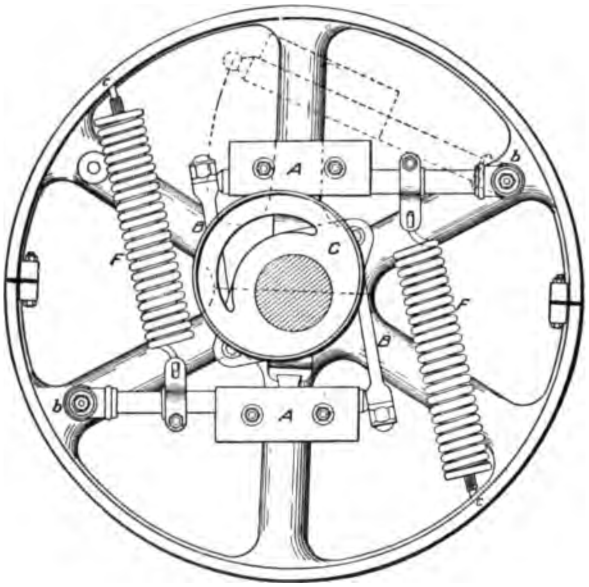

Fig. 42.

One cause of loss of power in the steam engine is friction. In all engines there are so many moving parts that it is of great importance that friction should be reduced as much as possible. This is done by making the surfaces in contact smooth and of ample size; also making them of different metals and using oils or other lubricants. The effect of the lubricant is to interpose a thin film between the surfaces. This prevents their coming into actual contact. If the oil is too thin or the pressure too great, the lubricant is squeezed out and the metal surfaces come in contact.

Thus we see that there are certain qualities which a lubricant must have. They are as follows:

The lubricant must be sufficiently fluid, so that it will not itself make the bearing run hard.

It must not be too fluid or it will be squeezed out from between the bearing surfaces. If this happens, the bearing will immediately begin to heat and cut. The heating will tighten the bearing, and will thus increase the pressure and the cutting.

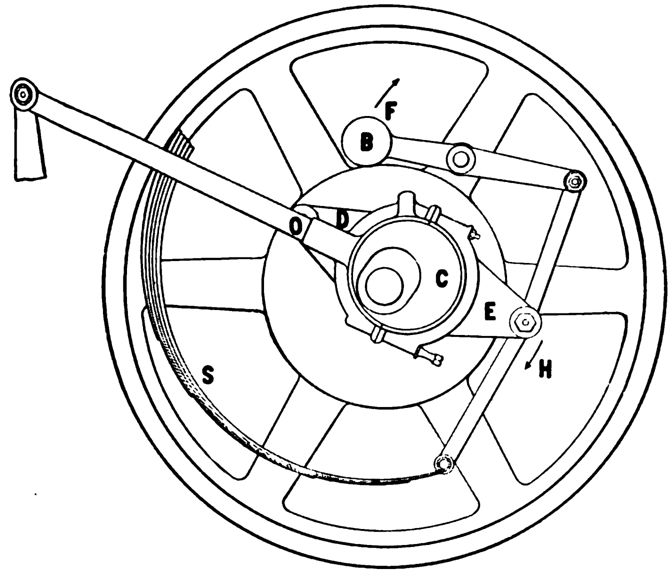

Fig. 43.

It must not gum or dry when exposed to the air.

It must not be easily decomposed by the heat generated. If it should be decomposed, it might form substances which would be injurious to the bearings.

It must not take fire easily.

It must contain no acid, and should form no acid in decomposing, as acids corrode the bearings.

Both *mineral* and *animal* oils are used as lubricants. Formerly animal oils were used entirely, but they were likely to decompose at high temperatures and form acids. It is important in using high-pressure steam to have "high-test oils," that is, oils which will not decompose or volatilize at the temperature of the steam. It was the difficulty of getting such oils which made great trouble when superheated steam was first used. Mineral oils will stand these temperatures very readily, and even if they do decompose, they form no acids.

The Liquid Lubricants, whether of animal, vegetable or mineral origin, may be used for ordinary bearings, but for valves and pistons heavy mineral oils only are suitable.

Solid Lubricants. *Graphite* is used as a lubricant. It is well adapted for heavy pressures when mixed with certain oils. It is especially valuable for heavy pressures and low velocities.

Metalline is a solid compound, containing graphite. It is made in the form of solid cylinders, which are fitted to holes drilled into the surface of the bearing. When a bearing is thus fitted no other lubricant is necessary.

Soapstone in the form of powder and mixed with oil or fat is sometimes used as a lubricant. Soap mixed with graphite or soapstone is often used where wood is in contact with wood or iron.

A preparation called *Fiber Graphite* is used for self-lubricating bearings. It is made of finely divided graphite mixed with fibers of wood. It is pressed in molds and afterward fitted to bearings.

For great pressure at slow speed, graphite, lard, tallow and other solid lubricants are suitable. If the pressure is great and the speed high, castor, sperm and heavy mineral oils are used.

For low pressure and high speed, olive, sperm, rape and refined petroleum give satisfaction.

In ordinary machinery, heavy mineral and vegetable oils and lard oil are good. The relative value of various lubricants depends upon the prevailing conditions. Oil that is suitable for one place might not flow freely enough for another.

The quality of oil is of great importance. In many branches of industry it is imperative that the machinery run as perfectly as possible. On this account and because of the high cost of machinery, only first-class oil should be used. The cylinder oil especially, should be high grade, because the valves, piston and piston rods are the most delicate parts of the engine.

Engines are lubricated by means of oil cups and wipers placed on the bearings wherever required. They are made in many forms, dependent upon the manufacturers. Commonly the oil cup is made with a tube extending up through the oil. A piece of lampwick or worsted leads from the oil in the cup to the tube. Capillary attraction causes the oil to flow continuously and drip down the tube. When not in use, the lampwick should be withdrawn.

The needle oil cup differs from the capillary oil cup in that a small wire or needle extends through the tube and oil; one end rests on the journal to be lubricated. The needle should fit the tube closely, so that when the machinery is at rest no oil will flow. When revolving, the shaft gives the needle a wabbling motion which makes the oil flow. To increase the supply, a smaller needle is used.

The oil cup shown in Fig. 44 is simple and economical. The opening of the valve is regulated by an adjustable stop. The oil may be seen as it flows drop by drop. The cylindrical portion is made of glass, so that the engineer can see how much oil there is in the cup without opening it.

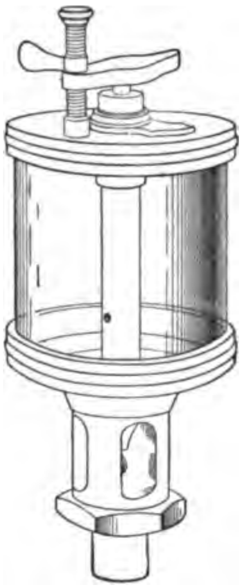
Fig. 44.

A form of wiper crank pin oiler is shown in Fig. 45. The oil cup is attached to a bracket. The oil drops from the cup into a sheet of wicking or wire cloth and is removed at each revolution of the crank pin by means of the cup which is attached to the end of the connecting rod.

Fig. 46 shows a centrifugal oiling device. The oil flows from the oil cup through the tube to the small hole in the crank pin by centrifugal force. It reaches the bearing surface by means of another small hole.

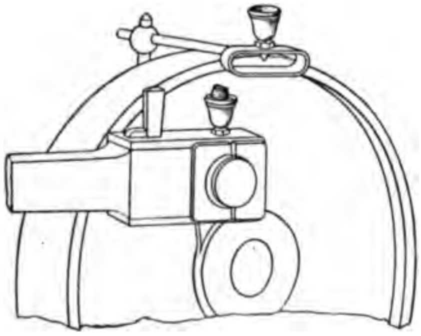

Fig. 45.

In oiling the valve chest and cylinder the lubricant must be introduced against the pressure of the steam. This can be done in several ways, in each of which it is introduced into the steam before it reaches the valve chest and is carried to the surfaces to be lubricated.

The oil can be forced into the steam pipe by a small hand pump or in large engines by an attachment from the engine itself. The supply of oil is, of course, intermittent if the pump is driven by hand, but continuous and economical if driven by the engine.

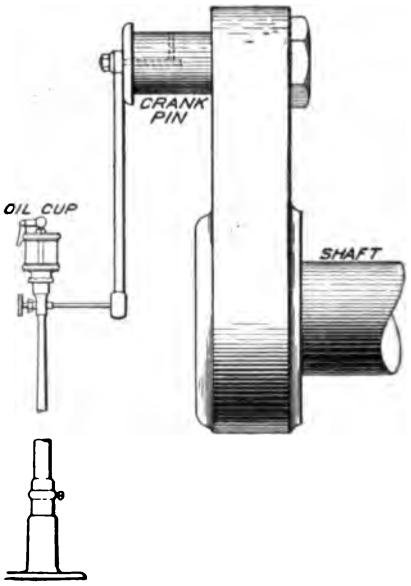

Fig. 46.

Sight Feed Lubricators. The most common device for feeding oil to the cylinder is that which introduces the oil drop by drop into the steam when it is in the steam pipe or steam chest. The oil becomes vaporized and lubricates all the internal surfaces of the engine.

Fig. 47 shows the section of a sight feed lubricator. The reservoir O is filled with oil. The pipe B, which connects with the steam pipe, is partly filled with condensed steam, which flows down the small curved pipe E to the bottom of the chamber O. A small portion of the oil is thus displaced and flows from the top of the reservoir O down the tube F, by the

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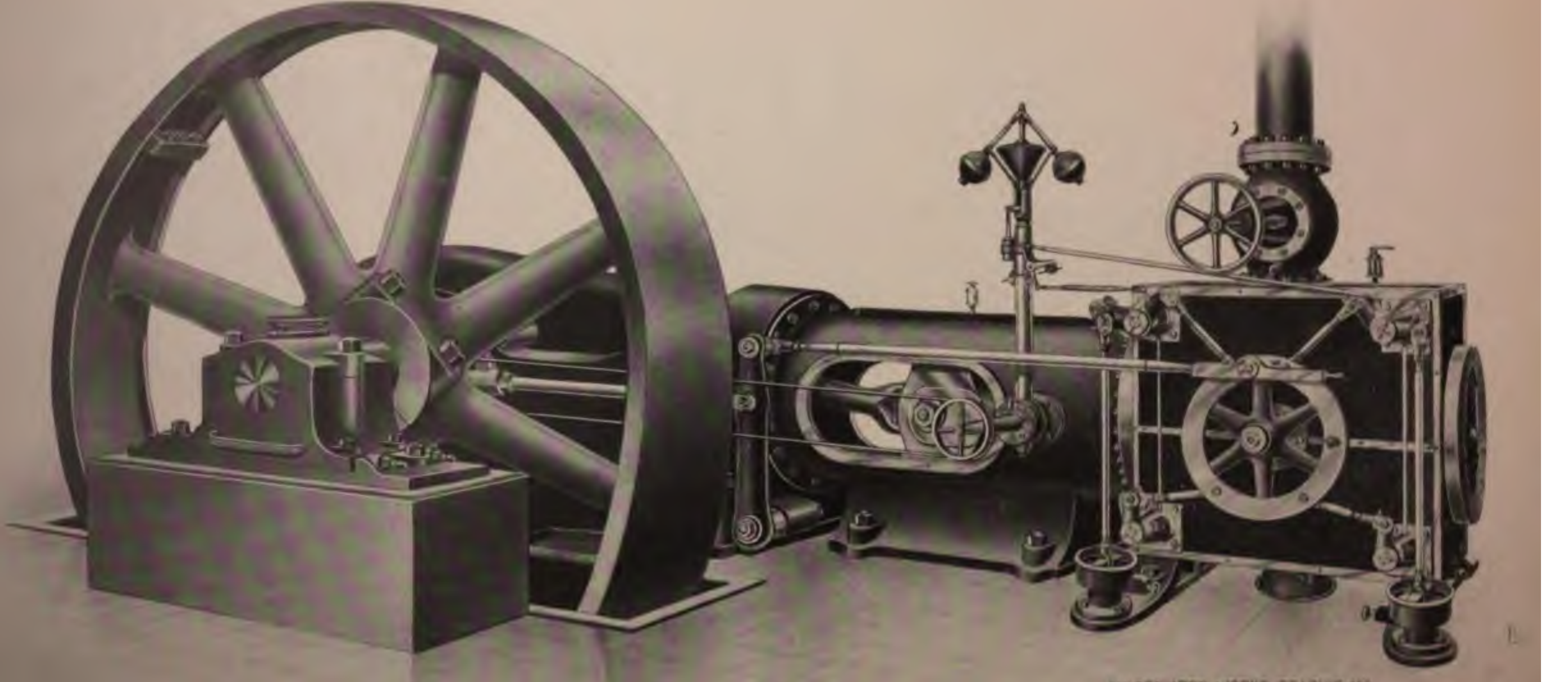


FULTON IRON WORKS, ST. LOUIS, MO.
BUILDERS.

regulating valve D, and up through the glass tube S, which is filled with water. It enters the main steam pipe through the connection A. The gage glass G indicates the height of water in the chamber O. To fill the lubricator, close the regulating valve D and the valve in pipe B; the oil chamber can then be drained and filled. If the glass S becomes clogged it may be cleaned by shutting valve D and opening the small valve H. This will allow steam to blow through the glass. After cleaning close valve H and allow glass S to become filled with water before opening the feed valve. The amount of oil fed to the cylinder can be regulated by opening D (Fig. 47) the proper amount. The exact quantity of oil necessary for the engine is not easily determined. For ordinary sizes it is between one drop in two minutes and two drops per minute.

Graphite is an excellent lubricant and can be introduced into the cylinder dry or mixed with some heavy grease. It has been used extensively because of the trouble which cylinder oil gives in the exhaust and in the boilers of condensing engines.

In slow-speed engines it is not hard to attend to the oiling; all the parts are moving slowly and can be readily examined and oiled. Many high-speed engines run so fast that it is impossible to examine the various parts, and special means must be provided for lubricating. It is specially important in high-speed engines that there should be no heating. High-speed engines are generally used for electric lighting, and it is absolutely essential that they be kept running at the required speed to avoid flickering lights. Thus, while there is greater liability to heating in high-

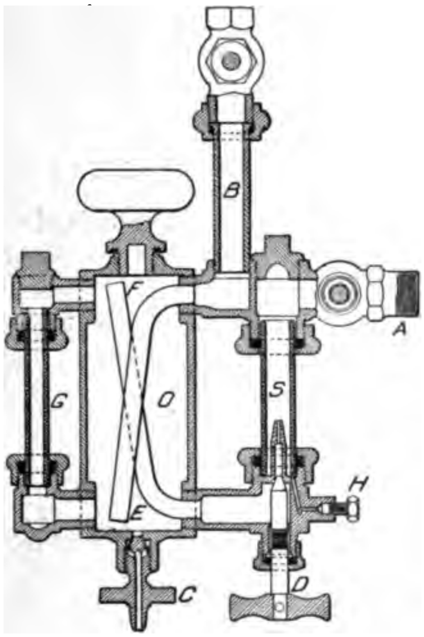


Fig. 47.

speed engines, there is also much greater loss in case heating compels the stopping of an engine.

In order to avoid the danger of forgetting to oil a bearing of a high-speed engine, it is customary to have all the bearings oiled from one place. All the oil is supplied to one reservoir and from this reservoir pipes lead to all bearings. If this is not done, large oil cups are supplied, as a rule, so that oiling need not be attended to as frequently.

In some high-speed engines the moving parts are enclosed and the crank runs in a bath of oil. This secures certain oiling and is very effective. All the bearings may be inside this crank case, so that all are oiled in this way. It is impossible for a careless engineer to overlook one point and so endanger the whole engine.

STEAM TURBINE.

The very earliest records of the steam engine describe a form of steam turbine. It consisted of a hollow sphere, as shown in Fig. 48, mounted on trunnions, through which steam was admitted to the interior. This steam escaped through pipes bent tangentially to the equator line of the sphere. The force of the escaping steam reacted upon the sphere, causing it to revolve on its trunnions. Many centuries later, in 1629, Branca, an Italian, invented a rotary engine (Fig. 49), in which a jet of steam struck the vanes of a wheel, and thus forced it around in much the same way that a jet of water acts on a Pelton water-wheel. These engines were of little, if any, practical value, and used an immense quantity of steam.

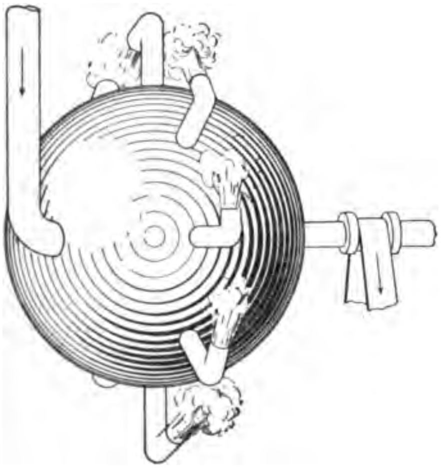


Fig. 48.

In 1705 the reciprocating engine was introduced, and by means of Watt's inventions became so efficient that the development of the rotary engine was out of the question. It will be remembered that Watt introduced the expansive use of steam in the reciprocating engine, which at this time could not be accomplished in the rotary engine, and until within the last few years practically nothing was done to develop the turbine.

Since the days of Watt there has been but one important thermodynamic improvement in the reciprocating engine; namely, the introduction of compound expansion. All other improvements have been in the nature of mechanical devices, and it seems reasonable to suppose that the greatest developments of the future may possibly be in the production of some type other than the reciprocating engine.

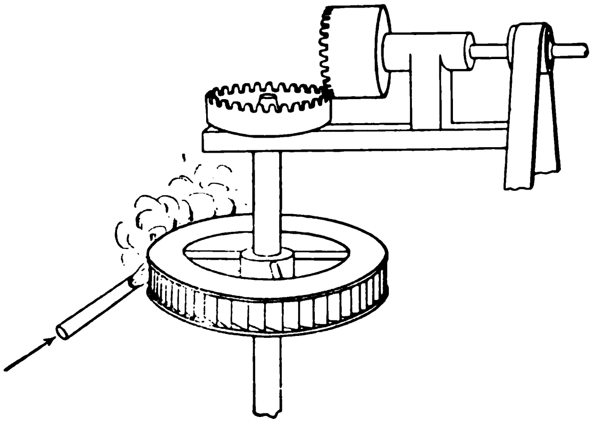


Fig. 49.

In 1883 De Laval invented a successful turbine for running a cream separator, and a short time later Parsons introduced another. Both of these engines employed the expansive force of steam, but each derived this force in a different way.

Since 1883 the development of the turbine has been very rapid. The first engine introduced by De Leval, although far ahead of the earlier forms, was still very wasteful of steam; but now such improvements have been made that their steam consumption compares very favorably with the consumption of good reciprocating engines.

A modern turbine is a tremendously high-speed engine. It does not derive its power from the static force of steam expanding behind a piston, as in the reciprocating engine, but in this case the expanding steam produces kinetic energy of the steam particles. These particles receive a high velocity by virtue of the expansion, and, acting upon the vanes of a wheel, force it around at a high speed of rotation in some such manner as a stream of water rotates a water-wheel.

In the reciprocating engine the expansion produces a force which presses on the piston. In the rotary engine the expansion produces velocity in a jet of steam. This is the fundamental difference between the two forms.

The essential principles of water turbines are equally true of steam turbines. The jet must strike the vanes without a sudden shock, and must leave them in another direction without any sharp deflections. For maximum efficiency the De Laval engine should have a jet velocity equal to one-half the linear velocity of a point on the wheel-rim; for the Parsons these velocities should be equal. If the velocity of steam is 8,000 feet per second, it is easily seen that even one-half of this would cause too great a speed of rotation for safety. It would be difficult to build a wheel that would be strong enough to withstand the centrifugal force at this high speed. It becomes necessary, therefore, to reduce the speed to the limits of safety, and run under a slightly less efficiency.

At such high speed the shaft and wheel should be perfectly balanced, in order that its center of gravity may exactly coincide with the axis of rotation. In practice it has been found impossible to balance the shaft perfectly; and in order that it may revolve about its center of gravity, various means are adopted to overcome the rigidity of an ordinary shaft and bearing. This makes high speed of rotation possible without any apparent vibration.

De Laval Turbine. The De Laval turbine shown in Fig. 50 consists of a wheel with suitably shaped buckets, against which a jet of steam is directed. The buckets are on the rim of the wheel and are surrounded by a casing B, which prevents the escape of the steam until it has done its work. A piece of this casing is cut away at A in order to show the buckets. The steam from the nozzle strikes these buckets and is deflected. Thus by the impact of the jet and the reaction due to its deflection, the wheel is caused to revolve at a high speed.

There are usually four nozzles that supply steam to the turbine, one of which is shown in section at D. These nozzles are small at the throat and diverge outward. By making them of the right length and with the proper amount of divergence, the steam can be expanded from the pressure of admission to the pressure of the condenser. Complete expansion is obtained in this diverging nozzle, and the steam leaves it at the exhaust pressure. The steam then works only by virtue of its high velocity.

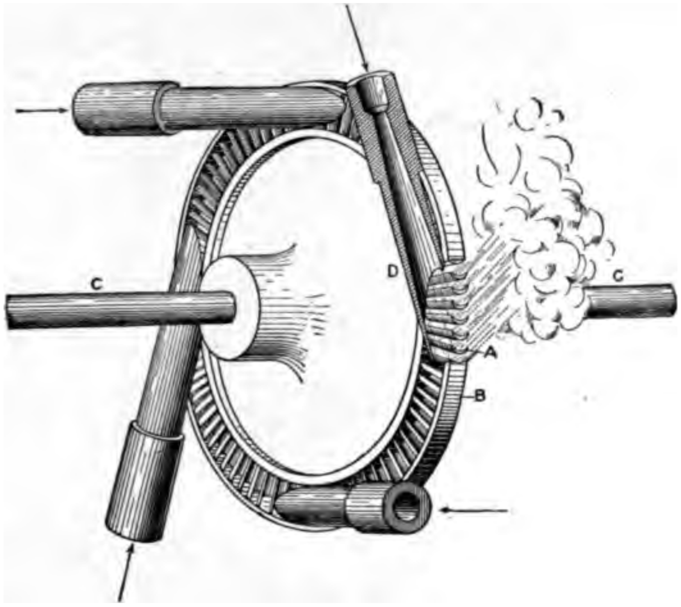


Fig. 50.

This turbine has a long, flexible shaft C which can deflect enough to make up for any eccentricity of the center of gravity of the shaft, and thus allow the shaft to revolve about its center of gravity and still have rigid bearings at the end.

Admission is regulated by a throttle-valve, controlled by a fly ball governor.

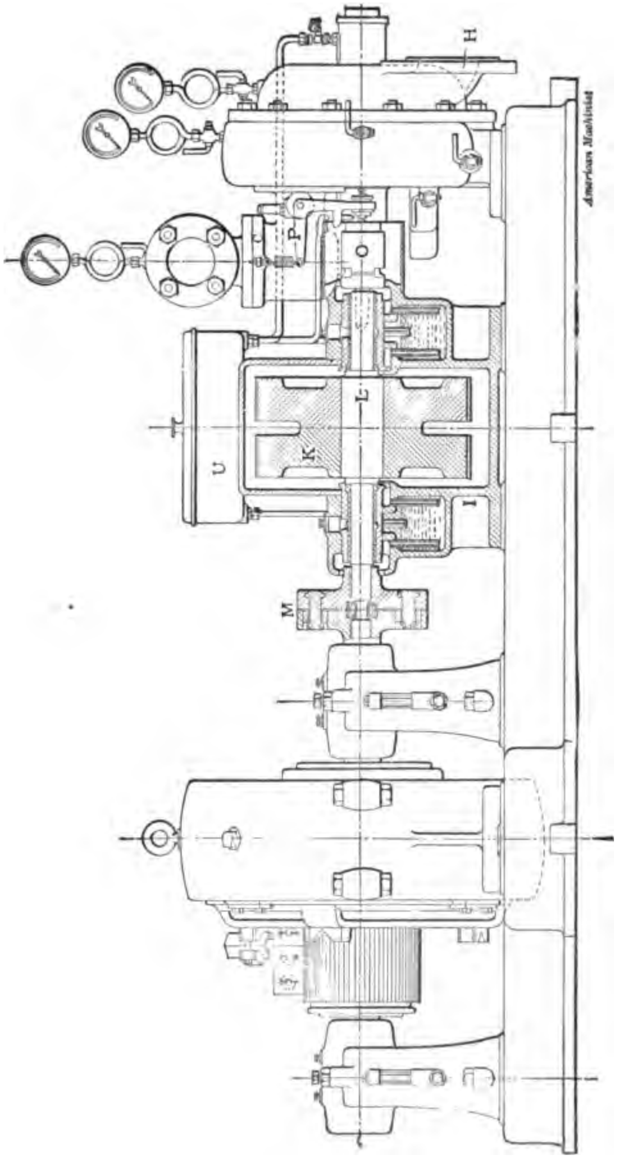


Fig. 51.

Fig. 51 shows a De Laval turbine connected with a generator.

The Parsons Turbine. Fig. 52 is a longitudinal section of a Westinghouse-Parsons turbine. Steam enters the chamber A and passes through the turbine vanes to the exhaust chamber B. The vanes are arranged as shown in Fig. 53, and consist of alternate sets, one stationary, the next movable. The steam strikes one and is deflected to the next; thus the action and the reaction occurring in rapid succession cause the movable sets, which are fixed to the shaft, to rotate at a high speed. As the steam passes the different sets of blades, the volume of the passages increases to correspond with the expansion of the steam. In the De Laval the steam was expanded entirely before reaching the wheel, but in the Parsons the expansion is accomplished in the engine itself. As the steam enters the chamber A (see Fig. 52) it presses on the turbine vanes and it also presses equally and in the opposite direction on C, which is really a piston fixed to the shaft. Thus we see that the pressure to the right is equal to the pressure to the left, and there is no end pressure on the bearing of the shaft. C_1 and C_2 balance the steam pressure in the chambers E and G. At H is a bearing which serves to maintain a correct adjustment of the balance pistons C. There is probably some escape of steam past these balancing pistons, but it is small. The exhaust steam at B presses the turbine toward the left, and would cause an end pressure on the bearing were it not that the pipe K opens a communication between the exhaust chamber B and the back of the balancing pistons, which makes the pressure equal at both ends.

The bearing consists of a gun-metal sleeve surrounded by three concentric tubes. There is a small clearance between these tubes which fills with oil and permits the bearing to run slightly eccentric to counteract any lack of balance in the shaft. Thus the shaft may revolve about its center of gravity, and this oil bearing serves the same purpose as the De Laval flexible shaft.

At P is shown a by-pass valve by means of which live steam may be admitted to the space E, if desirable. Of course this reduces one stage of the expansion, with a corresponding loss of economy, but will increase the power of the turbine. If the condenser fails on a condensing turbine it may still be run at full load by opening the by-pass valve.

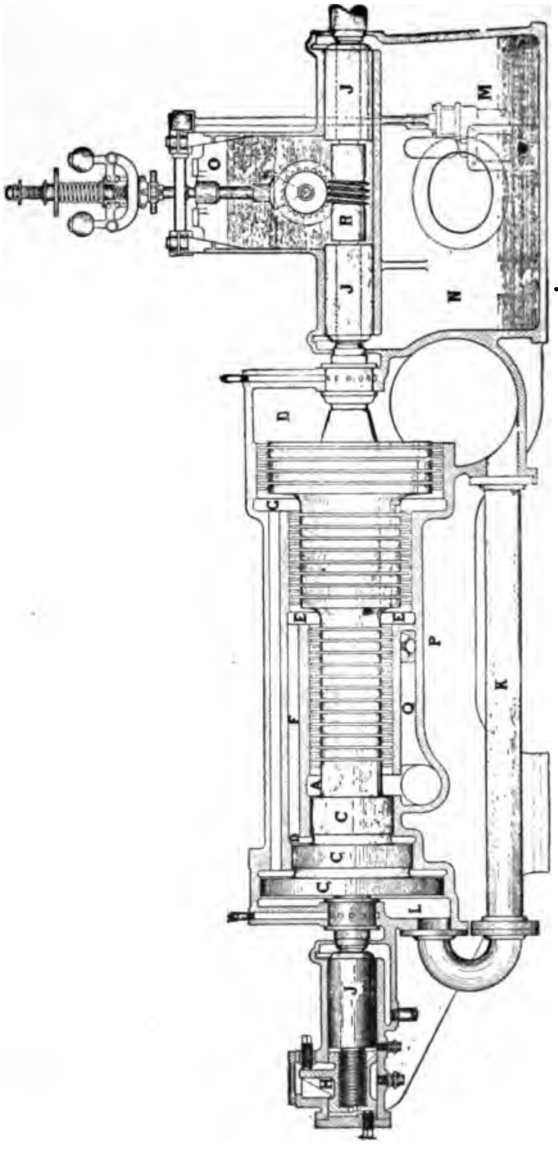


Fig. 52.

Steam is admitted to this turbine in puffs through a reciprocating valve. A fly-ball governor regulates the admission, which is always at boiler pressure.

For electric generators the turbine has many advantages, among them high speed and direct connection. They have small foundations and take up little space; there is slight loss from friction and few parts. Where slow speed is desired a reciprocating engine is probably the best.

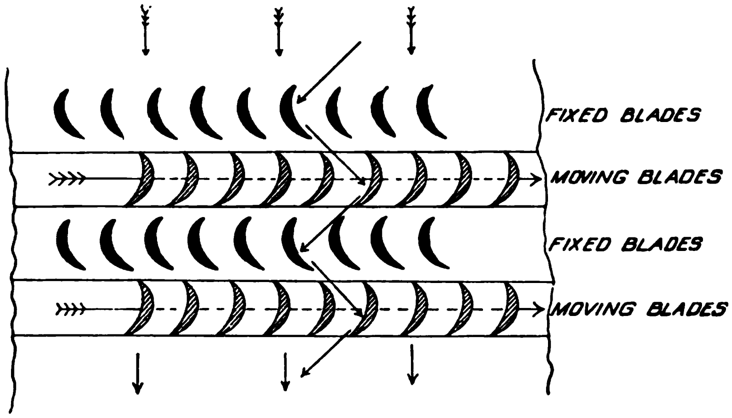
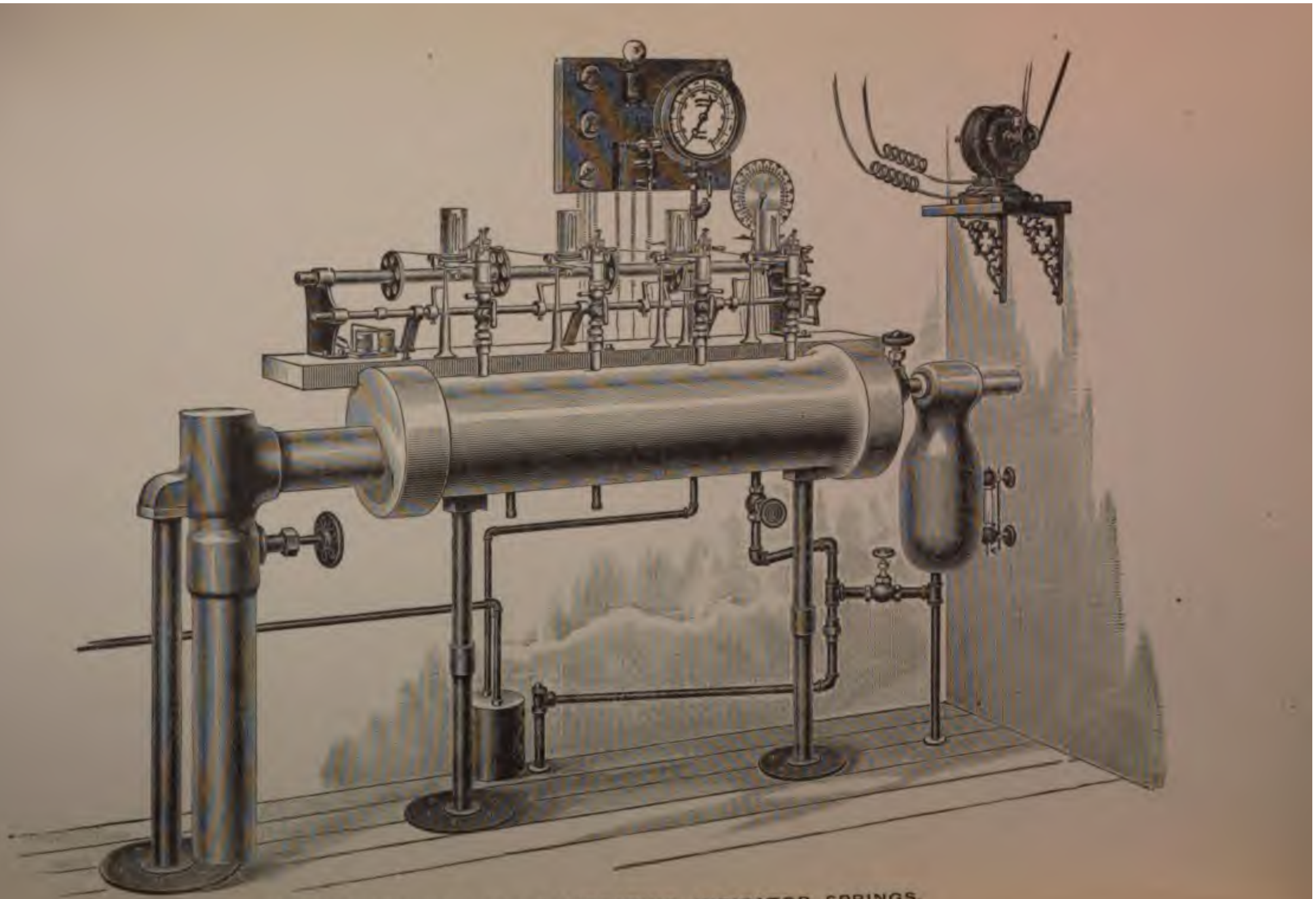


Fig. 53.



MERCURY COLUMN FOR CALIBRATING INDICATOR SPRINGS.

STEAM ENGINE INDICATORS.

A most important question concerning a steam engine is, "What is its horse-power?" or "How much work will it do in a given time?"

Work is defined as *pressure*, *force*, or *resistance* multiplied by the *distance* through which it acts.

Power is work done in a specified time.

In the steam engine, steam is the agent by means of which heat is transformed into mechanical work. It is the heat in the steam that does the work, not the steam itself.

Work is obtained from the heat in steam by confining it in a closed cylinder which is fitted with a piston and a piston-rod. Steam is admitted at one side of the piston while the other is open to the atmosphere or in communication with a condenser. The pressure of steam, usually 75 to 150 pounds per square inch, forces the piston to the other end of the cylinder, driving out the low-pressure steam in front of it. When it arrives at the other end, steam is admitted to that end and the piston is driven back.

The piston moves because the pressure on one side is greater than that on the other.

In order to move the piston, work must be performed. The amount of work is easily found, since work equals total pressure multiplied by the distance through which the piston moves.

Suppose a piston is 2 square feet in area and steam at a pressure of 64.7 pounds per square inch acts on it during the entire stroke of 4 feet; the other side of the piston being in communication with the atmosphere. The total pressure is then $2 \times 144 \times 64.7 = 18{,}633.6$ pounds. If this pressure acts through 4 feet it is evident from the definition that the work done per stroke will be,

$$18{,}633.6 \times 4 = 74{,}534.4 \text{ foot-pounds.}$$

Another method is as follows: The pressure on the above piston is $64.7 \times 144 = 9316.8$ pounds per square foot. The volume swept by the piston during one stroke is $2 \times 4 = 8$ cubic feet. If we multiply the pressure per square foot by the volume or 9316.8 by 8, we get 74534.4 foot pounds, the same result as before. Thus we see that work equals **unit pressure** multiplied by **volume.**

Let P = pressure on the piston in pounds per square foot.
p = pressure on the piston in pounds per square inch.
A = area of piston in square inches.
L = length of stroke in feet.
V = volume swept by piston in one stroke in cubic feet.
W = work done in foot-pounds.

Then from the above example,

Work = unit pressure multiplied by volume.

or, $W = P \times V$

It is evident that $P = 144\,p$, and $V = \frac{A}{144} \times L$.

Then we have these expressions for work,

$$W = P \times V = 144\,p \times V = 144\,p \times \frac{A}{144} \times L = p\,L\,A.$$

Suppose steam is admitted to the cylinder during the whole stroke, as in the above example, that is, one end of the cylinder is in communication with the boiler. The other end is open to the atmosphere. If we draw two lines at right angles to each other, as O Y and O X in Fig. 1, the volume of steam for any position may be represented by some distance measured on the line O X. Similarly the pressure of the steam at any position of the piston may be represented by the length of a vertical line parallel to the line O Y.

In the above example, the area of the piston was 2 square feet, the length of stroke 4 feet and the pressure by gage 50 pounds. Then we let O A = the atmospheric pressure = 14.7 pounds. At the beginning of the stroke the pressure (absolute) is $14.7 + 50 = 64.7$ pounds, represented by the distance O B, or A B = 50 pounds pressure. When the piston has passed through $\frac{1}{4}$ of the stroke it is represented as the point 1, or B 1 is the volume swept through when the piston has completed $\frac{1}{4}$ of the stroke. At this point the pressure is also 64.7 pounds as represented at 1′. Similarly, when the piston is at 2, 3, and 4 the corresponding pressure is 2′, 3′, 4′. Since the pressure is constant the line B D is parallel to O X. We see from the above that 50 pounds is the net pressure acting on the piston during the stroke, and is represented by A B and lines parallel to it. The volumes are represented by the horizontal line A C. Then since $W = P \times V$ it also equals $O B \times O X$ which is evidently the area of the rectangle O B D X. The area of the rectangle O B D X is proportional to the work done by the steam.

In Fig. 1, one inch on the line O Y = 40 pounds, then O B is 1.6175 inches long since it represents 64.7 pounds. Similarly O A must be .3675 inch since it represents 14.7 pounds. The line A C is 2 inches long; then referring to the preceding example, one inch in length $= \frac{8}{2} = 4$ cubic feet.

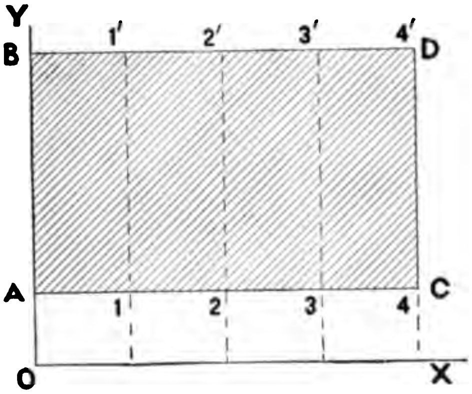


Fig. 1.

Since the rectangle O B D X is 1.6175 by 2 inches, the area is 3.235 square inches. But one inch in height equals 40 pounds pressure and one inch in length equals 2 cubic feet. Then $p\ V = 40 \times 3.235 \times 4 = 517.6$ foot-pounds and,

$$W = 144\ p\ V = 517.6 \times 144 = 74{,}534.4 \text{ foot-pounds.}$$

In the above cylinder the pressure acting on one side of the piston was 64.7 pounds per square inch. There was also a pressure of 14.7 pounds per square inch (the atmospheric pressure) acting in the opposite direction. Then the work done against the steam pressure is represented by the area O A C X and is equal to $144\ p\ V = 144 \times 14.7 \times 8 = 16934.4$ foot-pounds. Then since O B D X represents the total work done on one side of the piston and O A C X represents the work done *against* the piston the difference A B D C represents the **net work.** This net work is represented by the shaded area. Also if the amount of work done on the piston is 74,534.4 foot-pounds and the work done against the piston is 16,934.4 foot-pounds, the net work is the difference, or 57,600 foot-pounds.

In this theoretical discussion the same result may be obtained by subtracting the atmospheric pressure or back pressure from the absolute initial pressure and using the difference as the value of p. This value of p is called the mean effective pressure.

Then $64.7 - 14.7 = 50$ and

$W = 144\,p\,V = 144 \times 50 \times 8 = 57{,}600$ foot-pounds.

The area is proportional to the work done whatever the shape may be; provided the line B D represents the relation between pressures and volumes on the steam side of the piston and the lower line A C represents the relation between pressures and volumes on the exhaust side. If the engine is of the condensing type the line A C will be nearer O X, which is the line representing *absolute vacuum.*

Whatever the shape of the diagram, the area is equal to the area of a rectangle of the same length and a height equal to the mean height, or mean ordinate as it is called. The mean ordinate represents the mean or average net pressure on the steam side of the piston. Then we can follow these rules in finding the work of the steam from the diagram.

Multiply the area in square inches by the scale of pressures, by the scale of volumes and by 144, *or;*

Multiply the length of the mean ordinate by the scale of pressures, by the length of stroke, and this product by the area of the piston in square inches.

Example: The area of a diagram A B D C like that of Fig. 1 is 6.3 square inches and its length is 3 inches. The scale of pressure is 30 pounds per inch and the scale of volumes is 1.99985 cubic feet to the inch. If the piston is 20 inches in diameter and the length of stroke $2\frac{3}{4}$ feet, what is the work done per stroke?

Solution:

$W =$ area of diagram $\times$ scale of pressures $\times$ scale of volumes $\times$ 144.

$= 6.3 \times 30 \times 1.99985 \times 144 = 54{,}428$ foot-pounds.

$W =$ mean ordinate $\times$ scale of pressures $\times$ area of piston $\times$ length of stroke.

$= 2.1 \times 30 \times 314.159 \times 2\frac{3}{4} = 54{,}428$ foot-pounds.

Thus we see that we get the same result by both rules. The latter is the more common method because the mean ordinate is easily found and the scale of volumes seldom considered.

In our consideration of Fig. 1, steam was admitted to the cylinder during the *entire stroke*. In modern engines this method is rarely used; instead, steam is admitted during *part* of the stroke then the communication to the boiler is cut off, and the steam in the cylinder allowed to **expand**, as the piston moves forward, until it fills the entire volume of the cylinder. This is represented graphically in Fig. 2.

Steam is admitted to the cylinder until the piston reaches the point 2 which represents one-half the volume of the cylinder. Then the cylinder is half full of steam, that is, it contains $\frac{8}{2} = 4$ cubic feet. The four cubic feet of steam expand until they fill the cylinder. Since there is the same weight of steam present at every point in the stroke and the volume continued to increase, the pressure must diminish. This is shown in Fig. 2. The line B 2′ is horizontal because the pressure remains constant to the point of cut off. Then the pressure begins to fall as is represented by the curved line 2′ E. This curve is nearly an equilateral hyperbola.

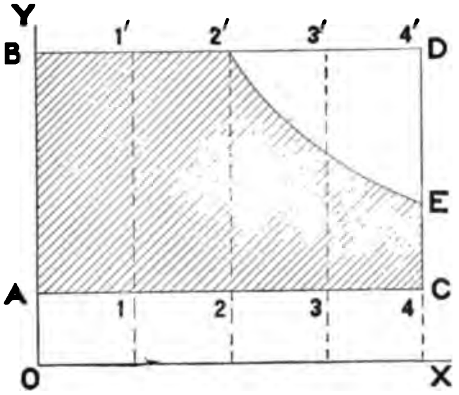


Fig. 2.

From Fig, 1 we know that the area B 2′ 2 A is proportional to the work done while the piston moves from A to 2 or during the first half of the stroke. If we use the same data as we did in Fig. 1, the work done must be one-half the work done in the first case, or $\frac{57600}{2} = 28800$ foot-pounds. Also the area B 2′ 2 A is easily found since it is a rectangle. The area 2′ E C 2 is found by dividing it up into small sections, by calculus or by the use of a planimeter.

It is easily seen that the area of the second case **Fig. 2**, is less than that of **Fig. 1**. Therefore the work done is less; but the amount of steam admitted is only one-half as much as in the first case.

In the first case, Fig. 1, 8 cubic feet of steam at 50 pounds pressure were admitted per stroke and the work done was found to be 57600 foot-pounds. In the second case only half as much steam is admitted and the work done is $\frac{57600}{2}$ + the amount represented by the area 2′ E C 2. Thus we see that there is a considerable gain by expanding the steam.

Watt's Diagram of Work. Fig. 8 illustrates the method adopted by James Watt to show the action of steam in the cylinder.

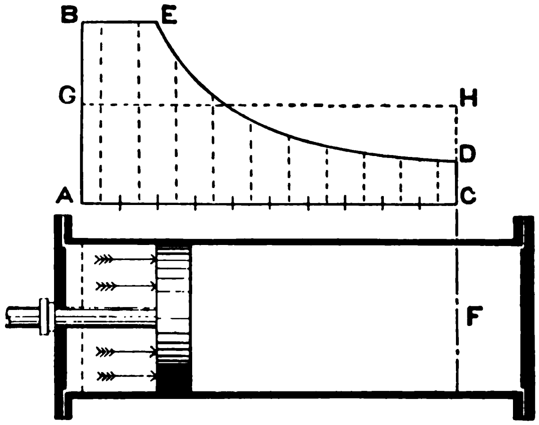
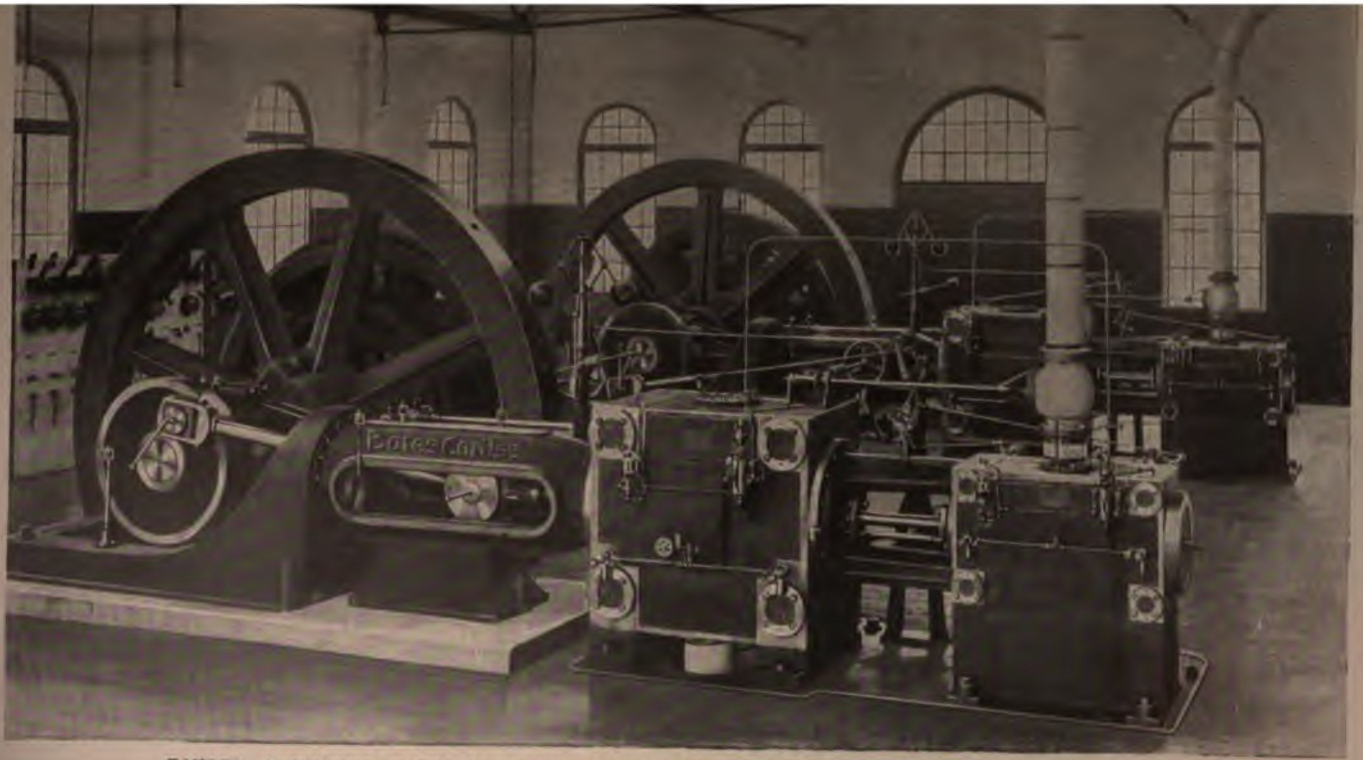


Fig. 8.

The horizontal line A C called the abscissa represents the length of the stroke and is divided into ten equal parts. The vertical line A B called the ordinate indicates the pressure of steam.

When the piston has moved to the point E steam is cut off, that is, a volume of steam equal to $\frac{1}{5}$ the volume of the cylinder expands until it fills the entire cylinder. The area may be found by adding the several pressures (shown by the dotted lines), dividing by the number of divisions, and multiplying by the length.

If by some arrangement of steam tight pistons working in cylinders and having pencils fastened to them, we could get a dia-



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gram like that shown in Fig. 3 it would be of great use but too large for convenience.

To obtain the same diagram on a small scale an indicator is used. The value of such a diagram has already been shown when finding the work done in the cylinder. The indicator has enabled engineers to bring the engine of today to its present state of excellence. A correct idea of the action of steam in the cylinder can be obtained only by means of an indicator. It shows whether or not the valves are set properly and how the condenser is working. It also shows the engineer which end of the cylinder is doing the most work. By comparing the expansion line with an equilateral hyperbola, with a curve of constant steam weight, or with an adiabatic curve for steam, the cylinder condensation is calculated.

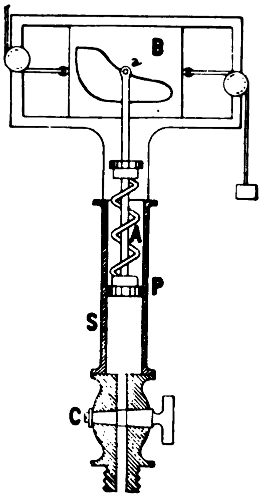

Fig. 4.

James Watt was the first to see the need of accurate knowledge of the action of steam in the cylinder. He invented the indicator. The improved form consisted of a steam cylinder S, about one inch in diameter and six inches long, in which a solid piston P, is accurately fitted. A spiral spring A, is attached to this piston, and controls the motion of a pencil *a*, which is also attached to the piston. This pencil can operate on a sheet of paper fastened to a sliding board, B. This board moves back and forth by means of a weight at one end and a cord at the other which is connected to some reciprocating part of the engine. The indicator cylinder S, may be put in communication with the engine cylinder by means of the cock C. With this instrument a complete diagram can be taken.

Watt's first indicator had no lateral motion, therefore all it showed was the pressure of steam in the cylinder and the perfection of the vacuum.

INDICATORS.

The diagram, or card as it is often called, obtained by the use of an indicator is the result of two motions. The *horizontal* movement of the paper corresponds exactly to the movement of the piston, and the *vertical* movement of the pencil is an exact ratio to that of the pressure of steam in the cylinder. The diagram represents by its *length* the stroke of the engine and by its *height* the steam pressure on the piston at the corresponding point of the stroke. The diagram shows the action of steam on one side of the piston only; to obtain the same information in regard to the other side it is necessary to take another diagram from the other end of the cylinder.

The essential features of an indicator are found in the instrument invented by James Watt. Since his time, however, the many improvements have made the indicator light, compact, durable, and accurate. Watt's diagram was traced on paper stretched on a sliding board but now a revolving drum is used. The height also of Watt's diagram was equal to the movement of the spring, and the pencil arrangement was a simple contrivance. In the indicators of the present day, the spring has a slight movement, the height of the card being obtained by a multiplying arrangement of levers. This method requires a parallel motion to obtain accuracy in the vertical lines; for if a lever is pivoted at one end and power applied near the pivot the lever tends to rise and the free end will describe an arc of a circle, not a straight vertical line.

THE THOMPSON INDICATOR.

Two views of the American Thompson Indicator, the outside and the inside, are shown in Figs. 5 and 6. The form of spring is shown in Fig. 7. The indicator consists of a cylinder in which a piston is fitted, a spring, multiplying lever and parallel motion for the pencil and a cylinder or drum for the paper. The piston, which is .798 inch in diameter = $\frac{1}{2}$ square inch in area, is fitted accurately to the cylinder and has a travel of about one-half inch. When the pressure of steam forces the piston upward it compresses the spring above it; the amount of compression varies with the strength of the spring. The rise of the piston causes the pencil to rise because of the system of levers. The cylinder to which the paper is attached rotates by means of a cord which is fastened to some part of the engine, the crosshead for example.

While the drum revolves and the steam pressure forces the piston to rise, the pencil, lightly touching the paper, decribes the diagram.

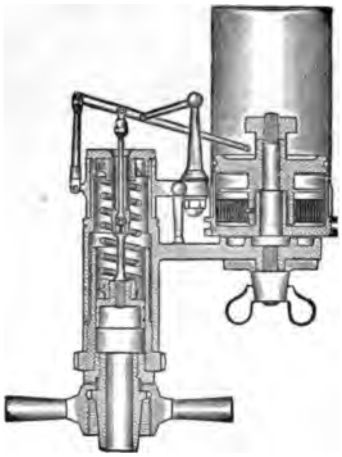

Fig. 5.

The parallel movement of this indicator is obtained by a link attached directly to the lever. It is so constructed that there is but little lost motion, hardly any friction and no appreciable error within the limited movement of the pencil.

The paper cylinder is so constructed that the tension of the coiled spring within the drum may be altered for different speeds of the engine. By this means the cord can be kept taut with little trouble. The cord is led through a hole and kept in contact with the scored wheel by another small one. By this means the cord can be run to any angle. It is convenient to have the cards of about the same size. If we used a spring of such a tension that the pencil would move one inch for every 60 pounds pressure, and there was but 30 pounds pressure in the engine cylinder, the diagram would be but one-half inch high. This diagram would be too small for accurate work. For this reason indicators are provided with sets of springs of varying stiffness which may be used according to the steam pressure. If a spring is of such stiffness that the pencil moves 1 inch for every 40 pounds pressure it is called a 40 pound spring. Others are called 10 pound, 20 pound, 30 pound, etc., springs.

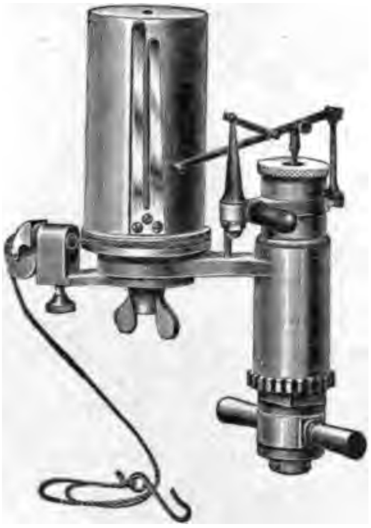

Fig. 6.

To Change the Springs. In selecting the spring for a given pressure, care should be taken that it will easily stand that pressure. A safe rule to follow: multiply the scale of the spring by $2\frac{1}{2}$ and subtract 15 for the vacuum.

For example: The maximum pressure for a 50 pound spring is 110 pounds, because $(50 \times 2\frac{1}{2}) - 15 = 125 - 15 = 110$. Springs are made in the following scale: 8, 10, 12, 16, 20, 24, 30, 32, 40, 48, 50, 56, 60, 64, 80, 100. For pressures from 70 to 90 pounds a 40* pound spring should be used, as 80 pounds pressure on a 40 pound spring will raise the pencil 2 inches, and this is a good height for the diagram.

If very high pressures are to be indicated, an extra piston, having an area of $\frac{1}{4}$ square inch, is used. This doubles the allowable pressure on the spring. For instance, if a spring can be used for 110 pounds pressure when the piston is $\frac{1}{2}$ inch in area, it can be used for 220 pounds pressure if the $\frac{1}{4}$ inch piston is used.

When the spring has been selected it is placed in the indicator. First unscrew the milled nut at the top of the steam cylinder and take out the piston with arm and connections. The pencil lever and piston are disconnected by unscrewing the small-headed screw which connects them. If a spring is connected to the piston it should be removed, the selected one substituted and the indicator put together. The spring should always be firmly screwed to the shoulder or the pencil will not properly indicate the pressure.

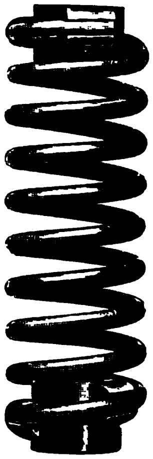

Fig. 7.

Care of the Indicator. Before attaching the indicator to the engine cylinder it should be taken apart, cleaned and oiled. If each part works freely and smoothly the spring may be put in and the indicator put together. After connecting to the cylinder, admit steam to it, but do not take cards until it is thoroughly warmed and blows dry steam through the relief. It is not necessary to use lead in connecting as it is likely to get into the indicator. After using, take the indicator apart, clean and oil. Only porpoise or fine watch oil should be used.

THE CROSBY INDICATOR.

The internal arrangement of the parts of the Crosby Indicator is shown in Fig. 8. The piston, 8, is formed with shallow channels on its outer surface to retain oil which prevents leakage and lubricates the piston.

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* **Note**: In practice it is better to use a somewhat stiffer spring.

The socket in the center of the piston is supported by a central web and projects both upward and downward. The upper portion is threaded inside to receive the lower end of the piston rod. It has a vertical slot which allows the ball bearing on the end of the spring to drop into a concave bearing on the upper end of the piston-screw, 9, which is screwed into the lower part of the socket.

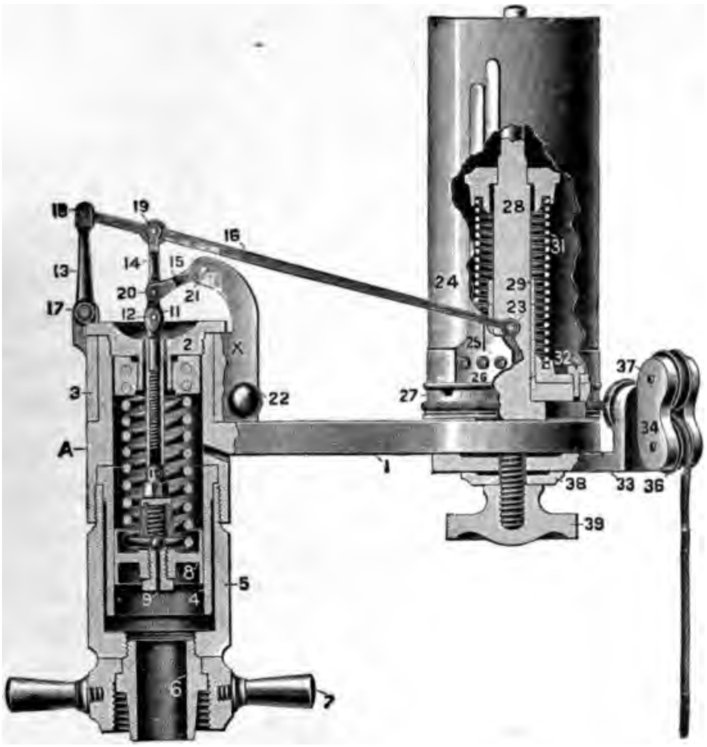

Fig. 8.

The piston-rod, 10, is made hollow, with the lower end threaded. When the piston-rod is connected to the socket, the former should be screwed into the socket as far as it will go.

The height of the atmospheric line on the diagram depends upon the amount the swivel head, 11, is screwed into the top of the piston-rod.

A small projection on the lower side of this cap is threaded to screw into the top of the spring and hold it firmly in place. The moving parts are kept in line by this cap. The pencil mechanism is supported by the sleeve 3, which surrounds the upper part of the cylinder; it turns freely and is held in position by the cap.

The pencil mechanism is made as light as is consistent with strength and stiffness. The pencil moves exactly parallel to the piston because the fulcrum of the mechanism and the point of attachment to the piston-rod are always in a straight line. The pencil point moves six times as far as the piston because of the multiplying levers.

The drum 24, is one and one-half inches in diameter, and is rewound when the string is pulled, by a short spiral spring 31.

The piston spring is made of a single piece of steel wire wound from the middle into a double coil. The ends are screwed into a brass head having four radial wings. At the bottom of the spring a small steel bead is firmly attached to the wire. This forms a ball and socket joint with the lower end of the piston-rod. This joint is light and allows the spring to yield to pressure from any direction. These springs are made in the following scale: 8, 12, 16, 20, 24, 30, 40, 50, 60, 80, 100, 120, 150, and 180.

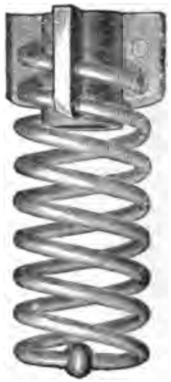

Fig. 9.

To Insert the Spring. First unscrew the cap 2, then lift the connected parts free from the cylinder by means of the sleeve. The hollow wrench should be held in an inverted position and the piston-rod inserted until the hexagonal part engages the wrench. Then, having the spring shown in Fig. 9 inverted, insert the combined wrench and piston-rod until the steel bead and the end of the spring rests in the concave seat. Now invert the piston and pass the transverse wire at the bottom of the spring through the slot until the threads at the bottom of the piston-rod engage those inside the socket of the piston. With the wrench screw it in as far as it will go.

The piston screw should be loosened slightly before the piston-rod is screwed in, and afterward set up against the head lightly to prevent lost motion. Then with the sleeve and cap upright, engage the threads of the swivel head with those inside the piston-rod and screw it up until the lower projection of the cap engages the threads inside the spring top; continue the process until the spring is screwed up firmly against the cap. Holding only by the sleeve 3, turn the piston and the connections until the top of the piston-rod is flush with the shoulder on the swivel head.

Now that the piston and all the connections are in their places, the whole may be inserted in the cylinder and the cap screwed down, which will fix all parts in their proper places. If there is a spring in the cylinder first detach by reversing the above process.

THE TABOR INDICATOR.

The Tabor Indicator is shown in Fig. 10. It is used extensively in the navy. The principle of action and details of construction are similar to the indicators already described; the chief peculiarity being the means employed to obtain a straight line movement for the pencil. Inside the steam cylinder is a lining in which the piston moves. This lining can expand when heated. In the side of the cylinder small holes allow any steam which may leak by the piston to escape. The piston-rod is connected at one end to the piston by means of a ball and socket joint; the other end is connected to the pencil mechanism.

The **pencil mechanism** consists of three pieces, the pencil lever, the back link, and the piston-rod link. The two links are parallel for every position of the pencil. Thus the lower pivots of these links and the pencil point are always in the same straight line. The straight line movement of the pencil is obtained by means of a curved slot in a stationary plate. The pencil bar is provided with a roller which is fitted in such a manner that it can roll from one end of the slot to the other. The curve of the slot guides the pencil bar and is of such a radius that the pencil is caused to move in a straight line. The curve compensates for the tendency of the pencil point to move in an arc of a circle. The pressure of the pencil on the paper is regulated by a screw which strikes a stop plate attached to the frame. The end of the pencil bar is formed for either a pencil lead or a metallic marking point.

The drum for the paper is made similar to those of other indicators. The backward movement is obtained by a flat spiral spring placed under the drum. The tension of this spring is altered by loosening a thumbscrew, lifting the carriage, and winding or unwinding. A simple pulley guides the driving cord in any direction.

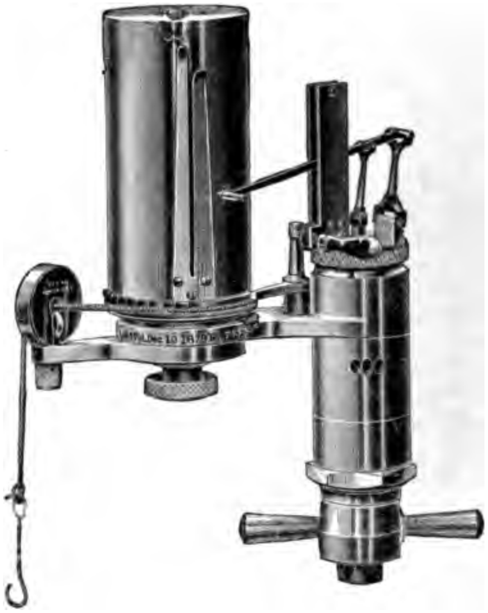

Fig. 10.

The indicator is attached by a coupling having a single thread.

The **springs** of the Tabor indicator are of the duplex type, that is, they are made of two spiral coils of wire. A 50 pound spring is shown in Fig. 11. The wire terminates in fittings at each end. The spring is attached to the upper side of the piston by means of threads cut on the inside of the fitting and on a projection on the piston. The top of the spring is attached to the under side of the cover in a similar manner. The springs are made in the following scales, 8, 10, 12, 16, 20, 24, 30, 32, 40, 48, 50, 60, 64, 80, and 100 pounds. The maximum safe steam pressures (absolute) to which these springs may be subjected are respectively, 10, 15, 20, 24, 40, 48, 70, 75, 95, 112, 120, 140, 152, 180, and 200 pounds.

Change of Location of Atmospheric Line. Unscrew the cap and lift the sleeve and connections from the cylinder. Then turn the piston to the left or right according as the pencil is to be raised or lowered. One revolution causes the pencil to rise ⅛ inch.

Care of the Indicator. Before attaching the indicator to the engine, steam should be blown through the pipes and cocks so that all particles of dust may be removed. After using, the indicator should be carefully wiped and oiled. The cylinder cap should be unscrewed and all the parts connected to the piston removed. The piston spring and piston-rod should be detached, carefully wiped dry, and then oiled. The inside of the cylinder also should be oiled. Then the piston and piston-rod should be placed in the cylinder and the spring placed in the box. If the indicator has not been used for some time the oil may have become gummed. It may be easily cleaned by wiping with a cloth saturated with naphtha or benzine. It must be oiled again before using. A good test that the indicator is in proper working order is to detach the spring and after replacing the piston and piston-rod, raise the pencil to the highest point. When allowed to fall it should descend to the lowest point freely. The pencil should always have a smooth fine point.

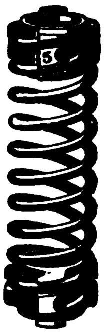

Fig. 11.

To Attach the Indicator to the Engine. Usually all first-class engines are prepared for the indicator before leaving the factory. Holes are drilled and tapped in the cylinder and have plugs screwed in them. These plugs are easily removed and the indicator connections screwed in.

When this is not the case, any engineer can perform the work. Before drilling the holes, in the cylinder, the heads should be removed so that the exact positions of the pistons and the size of the ports and passages may be known. Also with the heads off all chips and particles of dirt from drilling may be easily removed. If it is impossible to remove the heads, a little steam admitted to the cylinder just before the drilling is completed will blow the chips out.

Each end should be drilled and tapped for a one-half inch pipe thread. The holes must be drilled into the clearance space, so that the piston at the ends of the stroke will not cover them. They should also be placed so that currents of steam will not reach them. Before deciding just the points at which to drill the holes, it is well to consider every plan of indicating the engine. The type of engine, the position of the steam chest, the kind of crosshead, and the position of the eccentric and its connections should all be considered, as well as the most convenient place in the engine room. The holes should not be drilled until the plan shows the proper connections with the reducing motion, convenient access and free passage of steam to the indicator.

When the plan has been adopted, the engine should be placed on dead center, to determine the clearance. The holes should be drilled into the middle of the clearance space.

In common practice for horizontal engines the holes are drilled in the side of the cylinder at each end. Short half inch pipes with quarter upward bends into which the indicator coils may be screwed are inserted in these holes.

It may be more convenient to drill and tap into the top of the cylinder and attach the indicators directly.

For vertical engines, the upper head or cover and the side of the cylinder are often drilled and tapped for the upper and lower indicators respectively. It is preferable to connect the indicators to the sides because less pipe and fittings are required and better results obtained.

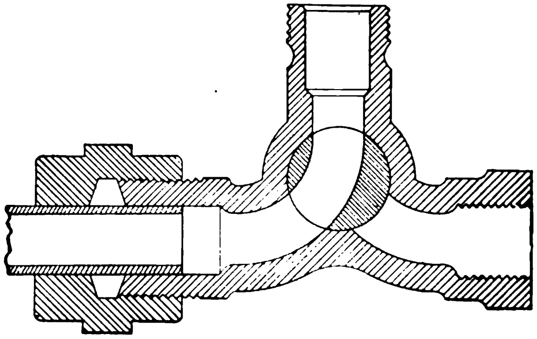

Fig. 12.

If only one indicator is to be used for both ends of the cylinder, it may be connected by side pipes and a **three way cock.** By this method both diagrams are taken on the same card and with the loss of but one revolution. Fig. 12 shows the section of a three way cock.

Reducing Motion. As we have already seen the length of the card represents the travel of the piston. As the length of card is obtained by the rotation of the drum the motion of the drum must be taken from some part of the engine which has a motion coinciding with that of the piston. The crosshead is the most common, reliable and convenient part. The length of the card is much less than the travel of the piston since the stroke is longer than the circumference of the drum so that the movement of the crosshead must be reduced to the length of the diagram.

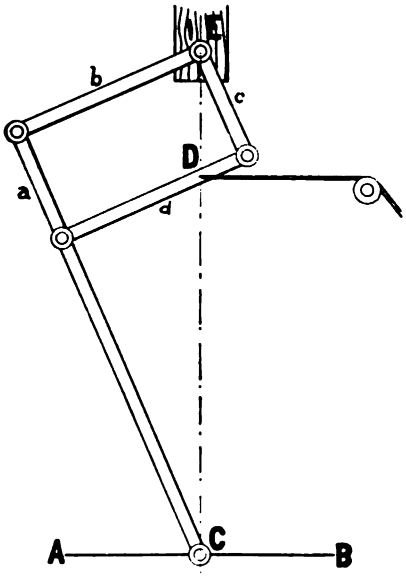

Fig. 13.

There are several devices employed to obtain this reduced motion.

A simple form of reducing motion, called the **pantograph,** is shown in Fig. 13. Four links, *a*, *b*, *c*, *d*, are joined in the form of a parallelogram. One link, *a*, is prolonged and pivoted at the crosshead C. The point where *b* and *c* join is pivoted at the fixed point E. The cord is fastened at D on the link *d*. The point D must be in the straight line connecting E and C. Then letting A B represent the stroke and *h* the length of the indicator diagram, we have

$$A\,B : h = E\,C : E\,D, \text{ from which}$$

$$E\,D = \frac{h \times E\,C}{A\,B}$$

Another form of pantograph is shown in Fig. 14. It is placed horizontally with the pivot, B, resting on a support opposite the crosshead when in mid-position. The pivot A, is attached to the crosshead; usually by having the stud A inserted in a hole drilled in the crosshead. If the pivot B is adjusted to the proper height and at the right distance from the crosshead, the cord from the indicator may be attached to the pin E without any pulleys. The length of the diagram is varied by adjusting the movable bar C D; the pin E must be in the straight line from A to B.

The pantograph is likely to become shaky and loose on account of its many joints. If well made it gives perfect motion.

The reducing motion shown in Fig. 15, called the Brumbo Pulley, is easily and quickly made and can be used on almost any engine. The wooden rod A is usually about twice as long as the stroke. It is pivoted by a bolt or screw at B, a fixed point. At the lower end it is connected by the wooden link, C, to the crosshead.

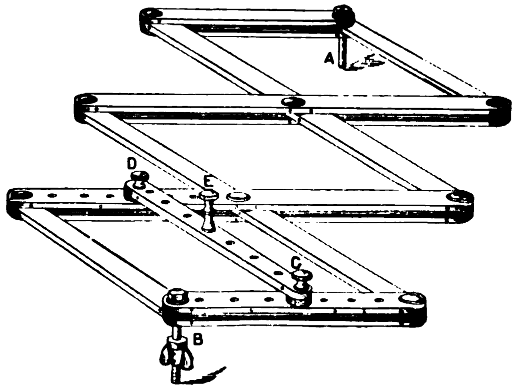

Fig. 14

This link C is usually about one-half the length of the stroke. The sector S may be made either of wood or metal. It should have a groove in the circular edge for the cord, and is made fast to the upper end of the lever A. Its center should coincide with that of the pivot B. The length of the radius of the sector may be found as follows. Divide the length of the lever by the length of the stroke, multiply the result by the length of the desired diagram and the product will be the radius of the sector. For instance, if the lever is 60 inches long and the stroke 30 inches and we wish the diagram to be 3 inches long, the sector should be 6 inches in radius for,

$$\frac{60}{30} \times 3 = 6.$$

To avoid the use of guide pulleys the lever should be hung so that it will swing in a vertical plane parallel with the guides and in line with the indicator. When the crosshead is at mid-stroke, the lever must be vertical and the point D must be below the axis of the cylinder because it comes above the line at the ends of the stroke.

The reducing lever used for large quick running engines is shown in Fig. 16. The rod A is made of pine wood, tapering toward the lower point and about one inch in thickness. The length is about one and one-half times the length of the stroke. It is suspended by a bolt or screw from some fixed point above the engine, and should swing edgewise and parallel to the guides of the crosshead. The steel stud at the bottom of the rod has a T shaped slot in an iron plate which is attached firmly to the crosshead. The slot should be long enough to retain the stud when the crosshead is at the end of the stroke. To find the point at which the indicator cord should be attached, divide the length of the lever by the length of the stroke of the piston and multiply the quotient by the length of the desired diagram. The product is the distance of the point from the pivot at the top of the lever.

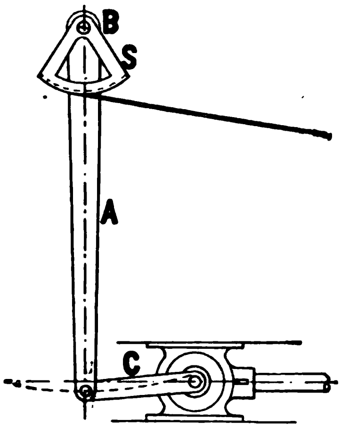

Fig. 15.

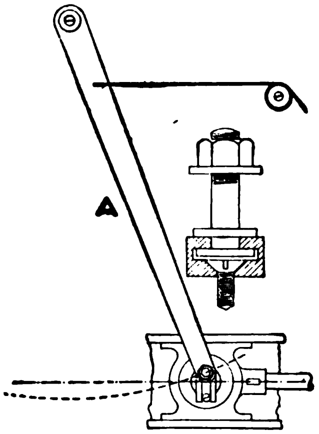

Fig. 16.

Example. A lever is 45 inches long, the piston stroke 30 inches and the diagram to be $3\frac{1}{4}$ inches long. At what distance from the pivot should the indicator cord be attached?

$$\frac{45}{30} \times 3\tfrac{1}{4} = 4\tfrac{7}{8} \text{ inches.}$$

Having placed the indicator in position and obtained the reducing motion, the length of the cord must be so adjusted that the drum will not strike the stops at either end of the stroke.

For convenience an approximate length of cord is first found and the cord cut in two parts, one attached to the reducing motion; the other to the indicator. A hook should be fastened to the free end of the piece attached to the indicator. A loop is made in the free end of the piece from the reducing motion. The hook is then attached to the loop and the extra length of cord taken up by tying knots. Another method for adjusting the length of the cord is the arrangement shown in Fig. 17. The hook A is attached to the indicator cord. The cord B from the reducing motion passes through the holes in the plate P as shown. To adjust the length of the cord it is slacked at the point B and the plate slipped along the cord.

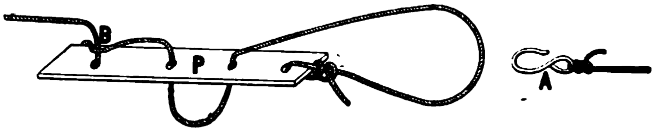

Fig. 17.

To Take Indicator Diagrams. The indicator should be in good working order before attaching to the cylinder; it should be clean, well oiled, and the levers and springs should work smoothly. The pencil point should be sharp and the pressure adjusted to make a distinct fine line.

The spring should be selected that will give a diagram 1½ to 2 inches in height. If the spring chosen is too light, the lines are likely to be wavy from the vibration of the pencil levers.

When the indicator is in position a satisfactory reducing motion obtained and the cord adjusted, the paper should be wrapped smoothly around the drum. The edges projecting over the clips should be folded back so that they will not touch the pencil lever. Before taking the card, allow the steam to enter the indicator and move the piston up and down until the parts have become thoroughly warmed. Then pass the pencil against the paper long enough to take the diagram. Some engineers allow the pencil to remain in contact with the paper during but one revolution of the engine; others trace the diagram two or more times. After the diagram is taken the cock is shut and without unhooking the cord the pencil is again pressed to the paper to take the atmospheric line. The cord is then disconnected and the card removed from the drum. The scale of spring, the dimensions and speed of the engine, the date, and all useful particulars are written on the cards.

If one indicator is used for both ends the three way cock shown in Fig. 12 is opened to admit steam from one end, the diagram taken; then opened for the other end and that diagram taken. Then the steam is shut off from both ends and the atmospheric line taken.

As has been said before, the indicator is of great importance to the engineer. It is used to find the indicated horse-power of the engine, and by comparison of the indicated horse-power with the brake horse-power, the **mechanical efficiency** is obtained. The indicator card shows several other things; the *time* and *manner* of the **four events** of the stroke, namely, the admission, cut-off, release, and compression. These four events make up what is called the **steam distribution.** It shows faults in the setting and working of the valves.

We have seen how the indicator diagram represents the net work done on the piston in one stroke.

Work is equal to pressure multiplied by the distance through which it acts. The *distance* is the length of stroke multiplied by the number of strokes per minute. The *pressure* is the *average net* pressure acting on the piston during the stroke. This average net pressure is called the **mean effective pressure.** If we know from the indicator card the mean effective pressure per square inch, we can find the total pressure by multiplying it by the area of the piston in square inches.

The distance per minute is equal to the length of stroke multiplied by the number of strokes.

Let P = mean effective pressure in pounds per square inch.
A = area of piston in square inches.
L = length of stroke in feet.
N = number of strokes per minute.

Then the work done per minute,

$$W = P\,L\,A\,N.$$

Since one horse-power is the rate of doing work when 33,000 foot-pounds of work are done per minute the **indicated horse-power** of an engine is obtained by means of the formula,

$$\text{H. P.} = \frac{P\,L\,A\,N}{33{,}000}$$

The length of the stroke, the area of the piston and the number of strokes are easily found. Then all that remains to be determined before the horse-power is calculated is the value of P, or the mean effective pressure.

Suppose one side of the piston is in communication with the boiler during the entire stroke, the mean pressure is then the boiler pressure. But if the supply of steam is cut off before the stroke is completed, the mean pressure will not equal the boiler pressure. In Fig. 1 we saw that the area of the shaded portion was equal to the length multiplied by the height. The area of a figure of any shape can be reduced to that of a rectangle having a length equal to the extreme length of the figure. Then whenever we know the area and length, we can find the height or mean height by dividing the area by the length.

Suppose a diagram like that shown in Fig. 2 has an area of $5\frac{1}{4}$ square inches and its extreme length is $3\frac{1}{2}$ inches; then the height is 5.25 divided by $3.5 = 1.5$ inches. Then with any indicator card the area of the diagram is equal to the area of a rectangle, the length of which is known and the height can easily be computed.

Suppose that we have taken a card and know that the mean height is $1\frac{1}{2}$ inches. In order to find the horse-power we must reduce the $1\frac{1}{2}$ inches to pounds pressure. If we multiply the height by the strength of the spring we get the desired result. For instance, if we had used a 30 pound spring (that is one which causes the pencil of the indicator to move one inch for every 30 pounds pressure in the cylinder) the height $1\frac{1}{2}$ inches would equal $30 \times 1\frac{1}{2} = 45$ pounds pressure.

Example. An indicator card has an area of 1.925 square inches and is 2.2 inches long. If a 60 pound spring is used what is the mean pressure?

The mean height equals $\frac{1.925}{2.2} = .875$ inch and

$$.875 \times 60 = 52.5 \text{ pounds.}$$

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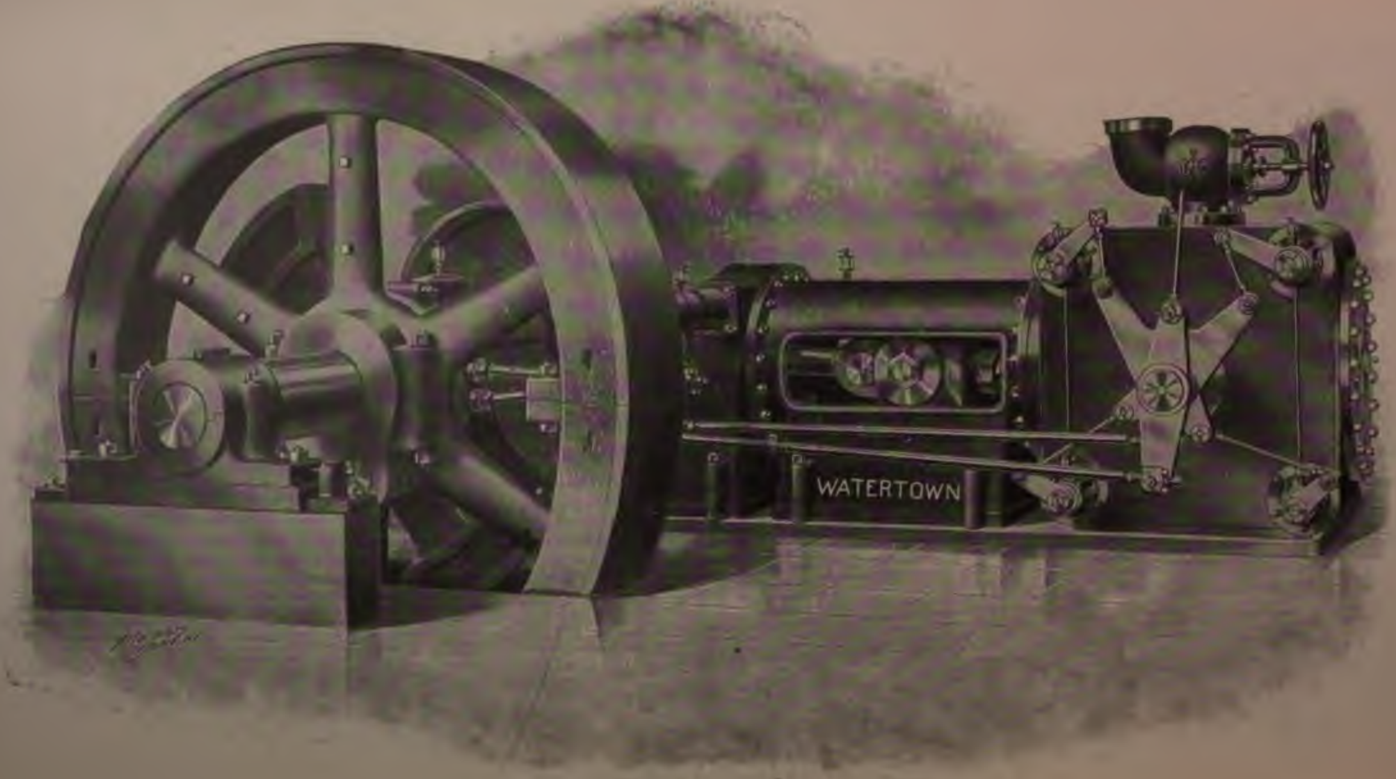


HIGH SPEED CORLISS ENGINE.
Watertown Engine Co.

Suppose the engine from which the above card was taken had a piston 14 inches in diameter and a stroke of 24 inches. If it were running at 150 revolutions per minute what is its horse-power? Assume the mean effective pressure to be the same for both sides of the piston.

$$\text{H. P.} = \frac{P\,L\,A\,N}{33{,}000}$$

$$= \frac{52.5 \times 2 \times 153.94 \times 300}{33{,}000}$$

$$= 147.\ \text{(about)}$$

In most engines more work is done at one end of the cylinder that at the other; it is not safe then to assume the mean effective pressure of one side the same as that of the other. Cards should be taken from each end and calculated for mean effective pressure separately, then averaged. Also the area of one side of the piston is greater than the other on account of the piston-rod. The two ends may be figured separately or the average area of the two sides of the piston may be used as the value of A.

Another method is to find the work done at each end of the cylinder and then add the results. This enables the engineer to know if his valves are set so that each end does about the same amount of work.

An engine has the following dimensions. The piston is 12 inches in diameter, the piston-rod is $2\frac{1}{8}$ inches in diameter and the length of stroke is 34 inches. While running at 92 revolutions cards were taken. The area of the card from the head end was 5.36 square inches, that of the crank end was 5.30 square inches, and a 40 pound spring was used. The cards were 3.72 inches long. We wish to know what horse-power the engine developed and which end was doing the most work.

The area of the piston is 113.097 square inches for the head end and 113.097 — 3.5466 = 109.55 square inches for the crank end.

Then for the head end,

$$\text{H. P.} = \frac{P\,L\,A\,N}{33{,}000} = \frac{P \times 34 \times 113.097 \times 92}{12 \times 33{,}000} = .8933\ P$$

and for the crank end,

$$\text{H. P.} = \frac{P\,L\,A\,N}{33{,}000} = \frac{P \times 34 \times 109.55 \times 92}{12 \times 33{,}000} = .8653\ P.$$

The value of P is found from the cards. Since the cards were 3.72 inches long and the area of the card from the head end was 5.36 square inches the mean ordinate is $\frac{5.36}{3.72} = 1.44$ inches which equals $1.44 \times 40 = 57.6$ pounds. The horse-power would be,

$$.8933 \times 57.6 = 51.45$$

For the crank end the mean effective pressure is found as before,

$$\frac{5.30}{3.72} = 1.425 \text{ and } 1.425 \times 40 = 57.$$

The horse-power would be,

$$.8653 \times 57 = 49.32$$

The horse-power is evidently the sum of these two quantities or $51.45 + 49.32 = 100.77$.

The head end is doing more work than the crank end but the difference is slight being only,

$$51.45 - 49.32 = 2.13 \text{ horse-power,}$$

or about 2.1 per cent. of the total power.

Considerable arithmetical work is necessary when the I. H. P. is found from the formula,

$$\text{I. H. P.} = \frac{\text{PLAN}}{33,000}$$

and the chances for error are of course, great. To save time and reduce the chance for error a table of engine constants has been prepared. The number of strokes, or twice the number of revolutions, multiplied by the length of stroke in feet is called the **piston speed**. Then in the formula $\text{I. H. P.} = \frac{\text{PLAN}}{33,000}$, L N = piston speed in feet per minute. In the following table, the I. H. P. of an engine is easily computed by multiplying the constant, corresponding to the diameter of the piston, by the piston speed and by the M. E. P. Or, in other words, the constants in the table equal the horse-power for an engine with a given diameter of piston having a piston speed of one foot per minute and a M. E. P. of one pound.

To Use the Table. If the diameter of the piston is an even number, the constant is found in the second column; if it contains a fraction the constant is found by following the column horizontally until the required fraction is reached. The constant multiplied by the piston speed in feet per minute and by the M. E. P. in pounds per square inch gives the I. H. P.

Example. An engine runs at 75 revolutions. The stroke is 4 feet; if the M. E. P. is 48 pounds and the piston $27\frac{5}{8}$ inches in diameter what is the I. H. P.

From the table the constant for a piston $27\frac{5}{8}$ inches in diameter is .0178355. The piston speed is 150 × 4 = 600 feet per minute. Then the I. H. P. is,

$$.0178355 \times 600 \times 48 = 513.66$$

The horse-power as above calculated is called the indicated horse-power and is usually written I. H. P. Although the above calculation shows the amount of power the engine develops it does not show the *available* power since part of the indicated horse-power is used to run the engine itself, that is, to overcome the friction of the parts. To determine how much power can be used to run machinery some form of absorption dynamometer or friction brake is attached to the engine. The power thus obtained is called the **Brake Horse Power** or B. H. P. It is more satisfactory for both the owner and builder to know the B. H. P. than to know the I. H. P.

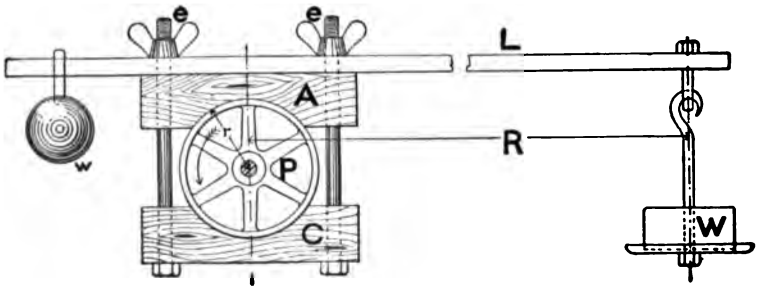


Fig. 18.

The Prony Brake, Fig. 18, is one of the simplest absorption dynamometers. The two wooden blocks A and C are held together against the rim of the pulley P by bolts. The thumb-nuts, *e*, *e*, being used to adjust the pressure. By means of the bolts the arm L is held to the upper block. From this arm is suspended the ball weight, w, which by sliding along the arm counterbalances the weight of the arm and pan at the other end. The pulley revolves at the required speed in the direction indicated by the arrow. The bolts are tightened until the lever remains stationary in a horizontal position when a known weight, W, is hung at the end.

The amount of work absorbed by the brake depends upon the weight W, the length, R, and the speed. It is independent of the diameter of the pulley and the pressure of the block because the moments of force about the center of the pulley are equal when the lever L, is horizontal. Letting f equal the co-efficient of friction, p the pressure of the blocks and r the radius of the pulley,

$$f\,p\,r = W\,R$$

The work done at the face of the pulley equals the force multiplied by the distance or the pressure multiplied by the number of feet passed through.

Let N = the number of revolutions per minute. Then the distance passed through per minute equals $2\,\pi\,r\,N$ and the work done equals $2\,\pi\,r\,N\,f\,p$. Then as $f\,p\,r = W\,R$, the work done at the rim of the pulley equals the left hand side of the equation multiplied by $2\,\pi\,N$, and to keep both sides equal we multiply W R by $2\,\pi\,N$. Hence the work done is expressed by the formula

$$2\,\pi\,N\,W\,R \text{ and,}$$

$$\text{B. H. P.} = \frac{2\,\pi\,N\,W\,R}{33{,}000}$$

$$= .0001904\,N\,W\,R$$

A Prony brake with an arm 4 feet long was attached to the pulley on the fly wheel of an engine. The weight in the scale pan was 50 pounds and the speed of the engine 300 revolutions. Find the brake horse power.

$$\text{B. H. P.} = .0001904 \times 300 \times 50 \times 4$$

$$= 11.424$$

The **rope brake** shown in Fig. 19 is easily constructed of material at hand and being self-adjusting needs no accurate fitting. For large powers the number of ropes may be increased. It is considered a most convenient and reliable brake. In Fig. 19 the spring balance, B, is shown in a horizontal position. This is not at all necessary; if convenient the vertical position may be used. The ropes are held to the pulley or fly-wheel face by blocks of wood, O. The weights at W may be replaced by a spring balance if desirable.

To calculate the Brake Horse Power, subtract the pull registered by the spring balance, B, from the load at W. The lever arm is the radius of the pulley plus $\frac{1}{2}$ the diameter of the rope. The formula is,

$$\text{B. H. P.} = \frac{2\,\pi\,R\,N\,(W - B)}{33{,}000}$$

$$= .0001904\;R\,N\,(W - B)\,^{*}$$

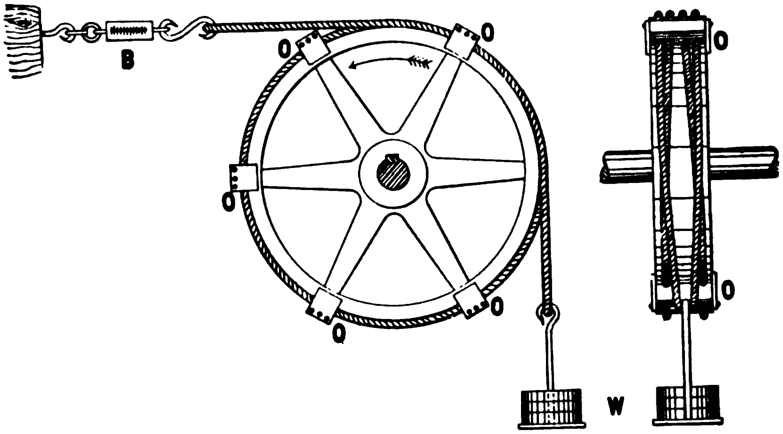


Fig. 19.

Example. A rope brake is attached to a gas engine. The average reading of the spring balance is 8 pounds. W = 80 pounds. If the radius of the brake wheel is 28 inches and the rope 1 inch in diameter, what is the B. H. P. when the engine makes 350 revolutions per minute?

$$R = 28 + \tfrac{1}{2} = 28\tfrac{1}{2}\text{ inches} = \tfrac{28.5}{12}\text{ feet}$$

$$\text{B. H. P} = .0001904\;R\,N\,(W - B)$$

$$= .0001904 \times \tfrac{28.5}{12} \times 72 \times 350$$

$$= 11.4\ \text{Ans.}$$

If both the indicated horse-power and the brake horse-power are known the power used in friction is found by subtracting the B. H. P. from the I. H. P.

---

* NOTE: If B is greater than W, the engine is running in the opposite direction. Use the formula B. H. P. = .0001904 R N (B — W).

The **mechanical efficiency** of the engine is the **ratio** of the B. H. P. to the I. H. P. or,

$$E = \frac{\text{B. H. P.}}{\text{I. H. P.}}$$

If an engine of 18.2 indicated horse-power develops at a trial 16.02 brake horse-power, what is its mechanical efficiency?

$$E = \frac{\text{B. H. P.}}{\text{I. H. P.}}$$

$$= \frac{16.02}{18.2} = .88$$

$$= 88\ \%\ \text{Efficiency.}$$

Brakes should be well lubricated. For small powers the heat generated by friction between the ropes or blocks and the rim of the wheel, will be conducted away by radiation but for large powers some additional means is necessary. In case there are flanges on the wheel, water can be introduced into the wheel, the flanges keeping it from flowing out and centrifugal force keeping it in contact with the rim. The amount of water can be regulated so that all may be evaporated, or a scoop can be arranged to carry off the water. In all cases the water should flow continuously.

To Find the Area of Cards. M. E. P. or the mean effective pressure is equal to the area of the indicator diagram divided by the length. The length is easily found by measurement but to find the area is more difficult since the shape is irregular. If the figure were regular its area could be found by geometry or by simple formulas.

The area of the indicator card can be found in two ways. By dividing the diagram into sections and by the use of a planimeter. The former is only an approximate method; the area thus found is nearly correct if the number of divisions is great.

Tangents at each end, perpendicular to the atmospheric line are first drawn. The horizontal distance between these tangents is then divided into 10 or more equal parts. The horizontal length of each section is then divided into two equal parts and lines perpendicular to the atmospheric line drawn through these points of division. The sum of the lengths of all these lines is divided by the number of lines to get the average. This average length or average ordinate multiplied by the scale of spring gives the mean effective pressure.

Fig. 20 is the card from the crank end of an engine. The line C L is the atmospheric line and the lines A D and E F are drawn perpendicular to it and tangent to the extreme ends of the diagram. The line A E is divided into 10 equal parts and lines are drawn through points marking the centers of the divisions. On each of these lines the length is marked. The sum of the lengths is 15.18 and 15.18 divided by 10 is 1.518. If the scale of spring is 40 pounds, 1.518 multiplied by 40 is the M. E. P. or 60.7 = M. E. P.

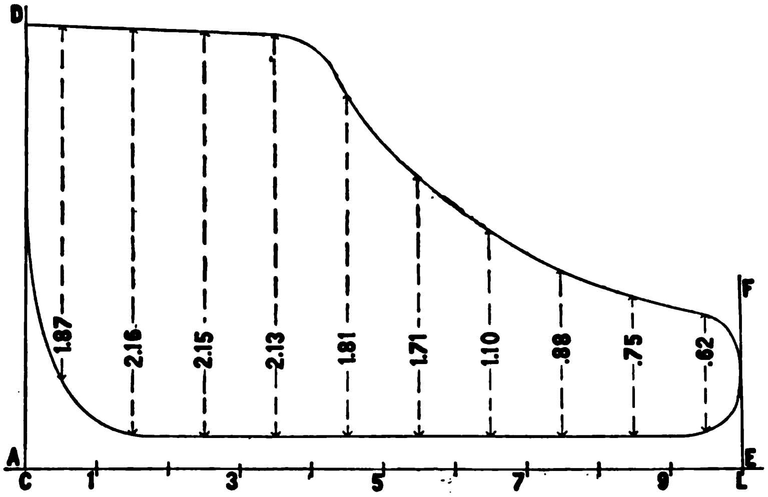


Fig. 20.

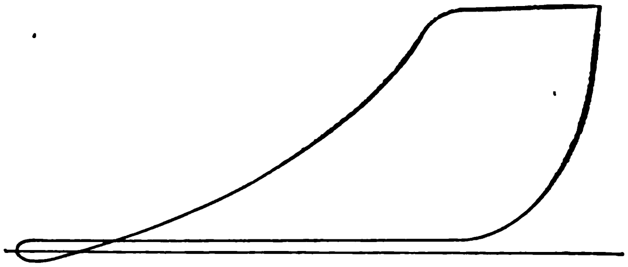


Fig. 21.

The horizontal length may be divided into any number of equal parts but 10 or 20 makes the computation easy. The operation of finding the M. E. P. for the head end is exactly the same. The average M. E. P. for one revolution of the engine is the average of the two mean effective pressures.

In case the diagram is very irregular it should be divided into 20 equal parts instead of 10. If there is a loop in the diagram as shown in Fig. 21 the area of the loop must be subtracted from the area of the other part as it represents work done by the piston on the steam and therefore loss.

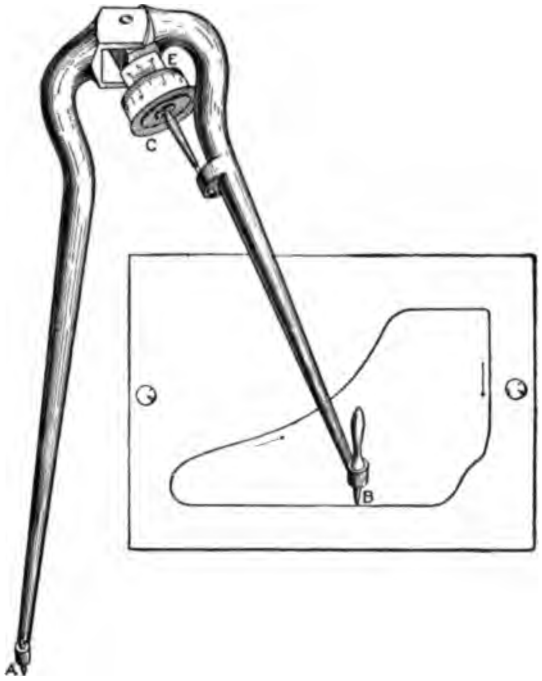


Fig. 22.

The lengths may be marked off on a piece of paper if a good scale is not at hand.

A more accurate result is obtained by using an instrument called the **planimeter.** There are several planimeters and averaging instruments in common use for determining the mean effective pressure of indicator cards. **The planimeter shown in Fig.**

22 is one of the most simple and is called the Amsler Polar Planimeter from its inventor Prof. Amsler. The cut is about one-half the size of the instrument. It consists of two arms free to move about a pivot and a roller graduated in inches and tenths of inches. A vernier is placed with the roller so the areas may be read in hundredths of a square inch. The point A is kept stationary and the tracer B is moved once around the outline of the diagram. The area in square inches of the diagram is read from the roller C and the vernier E.

To Use the Planimeter. The diagram should be fastened to some flat unglazed surface, such as a drawing board, by means of thumb tacks, springs or pins. The point A is pressed into the paper so that it will hold in place. The point B is set at any point in the outline of the diagram and the roller set at zero. Follow the outline of the diagram carefully in the direction of the hands of a watch as indicated by the arrows in Fig. 22 until the tracer has moved completely around the diagram. The result is then read to hundredths of an inch from the roller. Suppose after tracing over the outline we find that the largest figure that has passed the zero of the vernier is 3; the number of graduations (tenths) that have passed the zero to be 5 and the number (hundredths) of the graduations in the roller that exactly coincides with a graduation on the vernier to be 9. Then the area is 3.59 square inches.

Often at the start the roller is not adjusted so that the zeros coincide but the reading is taken and subtracted from the final reading. Thus if the first reading is 4.63 and the second 7.31 the area is 7.31 — 4.63 = 2.68 square inches. In case the second reading is less than the first, add 10 to the second reading then subtract.

This instrument is very valuable to an engineer who takes indicator cards. The results obtained are very accurate, the error being small. Ten or twelve diagrams can be measured by this instrument in the same time that is necessary to measure a single card by the method of ordinates.

It is well to run over the area three or four times and take an average as the tracing of the diagram cannot be absolutely correct at any time.

THERMAL EFFICIENCY.

The **thermal efficiency** of the steam engine is found in the same manner as that of any other heat engine. The efficiency depends upon the limits of temperature and not upon the nature of the working medium.

Let T_1 = absolute temperature of the heat received by the engine.
T_2 = absolute temperature of the heat rejected by the engine.
E = efficiency of engine.

Then,

$$E = \frac{T_1 - T_2}{T_1}$$

or, the efficiency equals the temperature of the heat rejected, subtracted from the temperature of the heat received and the result divided by the temperature of the heat received.

Suppose an engine is supplied with steam at 120 pounds absolute pressure and the exhaust is atmospheric pressure. What is the efficiency?

The absolute temperature corresponding to 120 pounds absolute pressure is $341.05° + 461°$ and the temperature of atmospheric pressure is $212° + 461°$.

Then,

$$E = \frac{802.05 - 673}{802.05} = .16 \text{ or } 16 \text{ per cent.}$$

If the engine had been of the condensing type and the exhaust pressure one pound above the vacuum, the efficiency would be as follows:

The temperature of one pound absolute pressure is $101.99° + 461°$.

$$E = \frac{802.05 - 562.99}{802.05} = .30 \text{ or } 30 \text{ per cent.}$$

In actual engines this efficiency cannot be obtained because the difference between the amount of heat received and that rejected is not all converted into work. Part of it is lost by radiation, conduction, leakage, etc. Also cylinder condensation reduces the efficiency.

The Theoretical Indicator Diagram. An indicator diagram is the result of two movements; a horizontal movement of the paper and a vertical movement of the pencil. The horizontal movement exactly corresponds to the movement of the piston of the engine and the vertical movement exactly corresponds to the pressure of steam in the cylinder.

The shape of the indicator card depends upon the manner in which steam is admitted to and released from the cylinder. Different engines give different shaped indicator cards and the cards taken from an engine vary with the conditions. Figs. 1 and 2 show theoretical indicator cards from a non-condensing engine without clearance; the former being for the case that has admission during the whole stroke. The diagram of Fig. 2 shows the cut off at $\frac{1}{2}$ stroke. All practical engines have clearance and slight compression; so the theoretical diagram assumes the shape shown in Fig. 23. In this card the admission line H A is vertical, the steam line A C is horizontal, the expansion line C D an hyperbolic curve, the exhaust line D B vertical, the back pressure line B F horizontal and the compression curve an hyperbola. The **actual** shape is somewhat different from the **theoretical** mainly because the valves do not open and close quickly, the ports offer some resistance to the passage of the steam and the back pressure is neither atmospheric in the non-condensing engine nor absolute vacuum in the condensing engine.

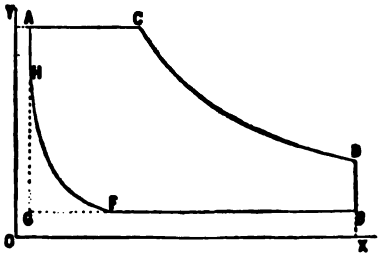


Fig. 23.

The diagram shown in Fig. 24 is a practical diagram and like those taken from engines.

The **atmospheric line** L M is the line drawn by the pencil of the indicator when the connection to the engine is closed and both sides of the piston of the indicator are open to the atmosphere. It is the zero of the steam gage.

The **admission line** H A shows the rise of pressure due to the admission of steam to the cylinder. If the steam is admitted quickly when the engine is nearly on dead center this line will be very nearly vertical.

The **steam line** A C is drawn while the valve admits steam to the cylinder. This line is horizontal if there is no wire-drawing.

The **point of cut off** C, indicates the point at which the admission of steam is stopped by the closing of the valve. This point is rounding since the valve closes slowly. Sometimes it is difficult to determine the exact point where cut off takes place; it is usually where the curve changes from concave to convex.

The **expansion curve** C D shows the fall in pressure as the steam expands while the piston moves toward the end of the stroke.

The **point of release** D shows the point at which the exhaust valve opens. The rounding is due to the slow action of the valve when opening. Because of this slow action of the valve, release begins a little before the end of the forward stroke.

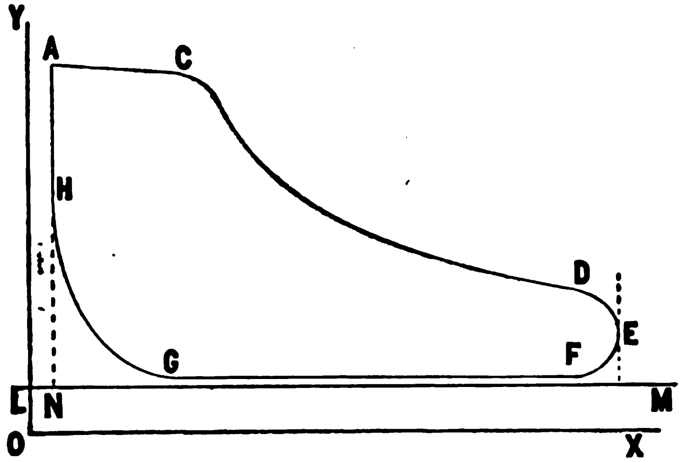


Fig. 24.

The **exhaust line** D E F represents the loss in pressure which occurs while the valve opens to exhaust at and near the end of the stroke.

The **back pressure line** F G shows the back pressure against which the piston acts during the return stroke. For a condensing engine this line is below the atmospheric line L M, the distance below being dependent upon the state of the vacuum in the condenser. For cards taken from a non-condensing engine the back pressure line is a little above the atmospheric line.

The **point of exhaust closure** G is the point where the valve closes to exhaust. The exact point is not clearly defined as the curve shows a change of pressure due to the gradual closing of the valve.

The **compression curve** G H shows the rise of pressure due to the compression of the steam remaining in the cylinder after the valve has closed to exhaust.

The **zero line** of pressure or line of absolute vacuum O X is drawn below and parallel to the atmospheric line. The distance between the lines O X and L M represents 14.7 pounds pressure.

The **clearance line** O Y is drawn perpendicular to the line of absolute vacuum and at a distance from the end of the diagram equal to the same per cent. of the length of the diagram as the clearance volume is of the piston displacement, or

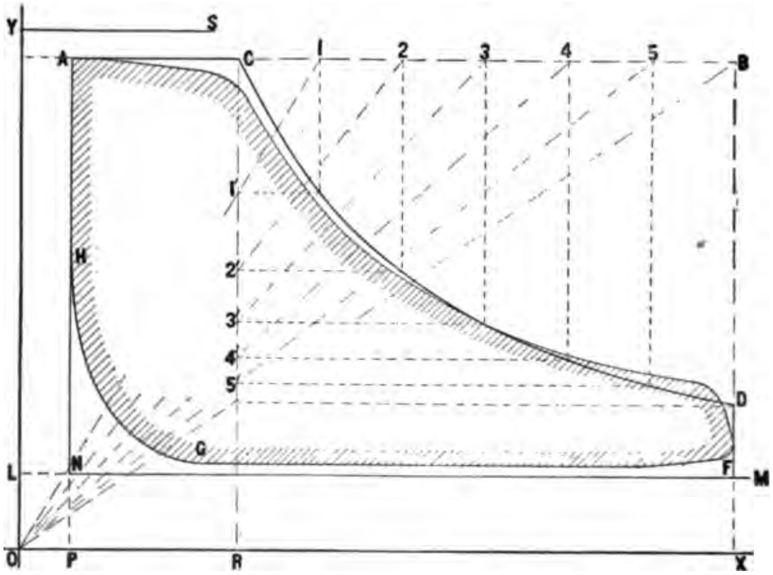


Fig. 25.

$$\frac{L\ N}{L\ M} = \frac{\text{clearance volume}}{\text{volume of cylinder}}.$$

It is readily seen that the area of an actual indicator diagram is less than that of a theoretical card. This is because of the round corners at cut off and exhaust, the back pressure and the compression. Sometimes it is useful, especially in designing engines, to draw the theoretical indicator card.

To Draw the Theoretical Card. To draw an ideal diagram draw P X equal to the length of stroke and O P equal to the clearance shown in Fig. 25. Draw O Y and P A perpendicular to O X and draw Y S parallel to O X and at a height corresponding to the boiler pressure.

The line of initial pressure A C is then drawn parallel to Y S and is usually taken as from 90 to 95 per cent. of the boiler pressure if there is no special cause for loss. Then take A C as the portion of the stroke at which steam is admitted so that $\frac{PX}{AC}$ = the ratio of expansion. The expansion line is considered a hyperbolic curve with O Y and O X as asymptotes. To draw the hyperbolic curve. First draw the line A C B parallel to the atmospheric line and F D B and R C perpendicular to it. Then make points 1, 2, 3, 4, etc., on C B and connect them with the point O. At the points 1′, 2′, 3′, 4′, etc., where these lines intersect the line R C draw parallels to C B until they meet perpendiculars from 1, 2, 3, 4. The point of intersection of these lines are points in the hyperbolic curve C D, as shown in Fig. 25. Any number of points may be used; but there must be enough to determine the curve.

The area A C D F N H is the theoretical card, with a given boiler pressure, ratio of expansion and an assumed back pressure. The actual card for the same data would be more nearly like the shaded area which lies mostly within the outline of the theoretical card. In designing engines it is well to know the ratio of the actual to the ideal card for all types of engines.

This ratio varies with the speed, type of engine, and kind of valves and has the following values.

For ordinary plain slide valve engines,	.8 to .9
For small engines, about,	.85
For large engines, about,	.90
For engines with high speed, about,	.75
For compound engines,	.75 to .80
For compound engines, high speed,	.65 to .75
For triple expansion engines,	.60 to .70
For triple expansion engines, high speed,	.50 to .60

CARDS FOR COMPOUND ENGINES.

In Fig. 26 the ideal cards of a **tandem** compound engine of the Woolf type are shown. The diagram A C D E F G H is from the high pressure cylinder and the diagram L M N P Q R from the low pressure. S T is the atmospheric line.

The line H A is the admission line, A C the steam line, and C D the expansion line. These lines are similar to those of a single cylinder engine diagram. At D, the point of release, the pressure drops slightly as steam is admitted to the low pressure cylinder. The back pressure line E F, of the high pressure cylinder is parallel to the steam line L M, of the low pressure cylinder. They would coincide if there were no resistance to the flow of steam and no heat loss. The flow of steam from the high to the low pressure cylinder is cut off at F and the steam remaining in the high pressure cylinder is compressed in the cylinder and in the pipes. The exhaust closes at G and steam is compressed in the small cylinder from G to H.

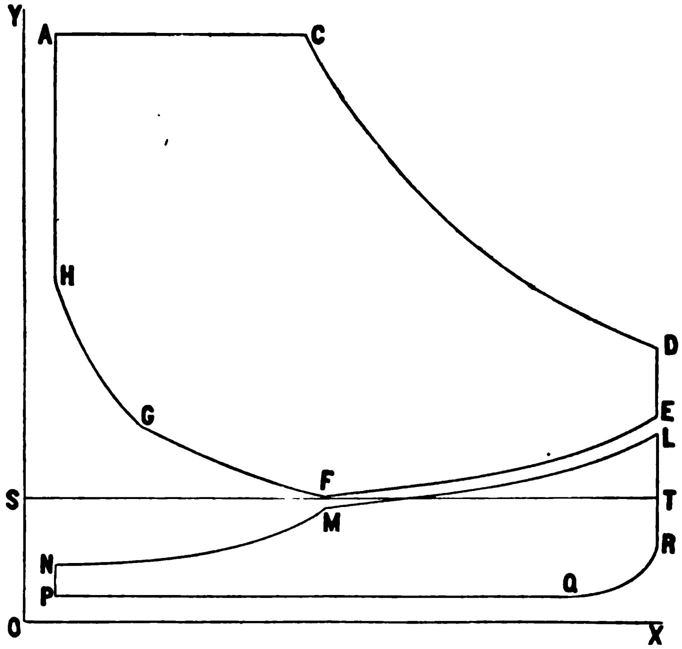
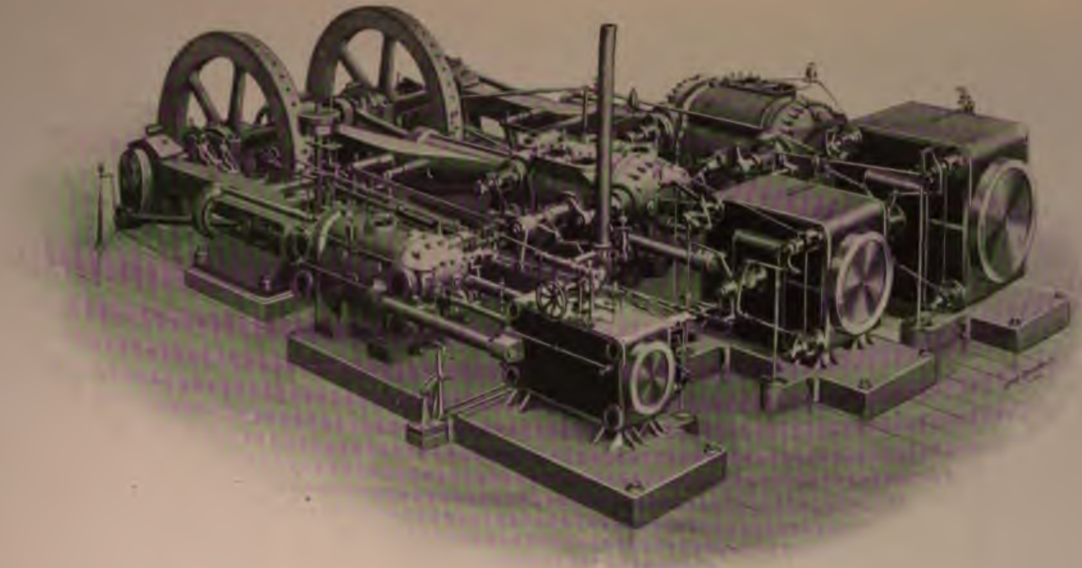


Fig. 26.



GENERAL VIEW OF THREE-STAGE TRIPLE EXPANSION AIR COMPRESSOR.
Nordberg Manufacturing Co.

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ASTOR, LENOX
TILDEN FOUNDATIONS

Cut off occurs in the low pressure cylinder at M, and the steam in the cylinder expands, the curve M N being an equilateral hyperbola. Release takes place at N and the pressure falls to that of the condenser. The back pressure line P Q and the compression curve Q R are like those of a single engine.

The diagrams shown in Fig. 27 are from a Woolf engine. The lines are similar to those of the theoretical diagram shown in Fig. 26.

With compound engines of the Woolf type the steam passes *directly* from the high pressure cylinder to the low pressure. In the **cross** compound, the cranks being at 90°, the piston in one cylinder is at the *end* of the stroke when the other is at the *middle* of the stroke. Therefore, a **receiver** must be used. The ideal cards from a cross compound engine are shown in Fig. 28. The dimensions are as follows:

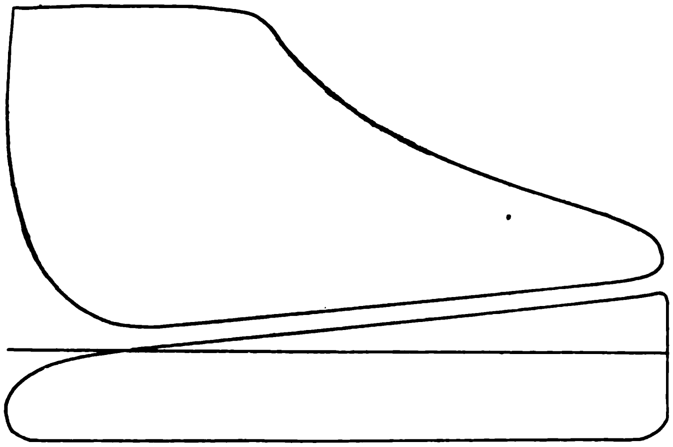

Fig. 27.

Volume of high pressure cylinder	=	5 cu. ft.
Volume of receiver	=	5 cu. ft.
Volume of low pressure cylinder	=	15 cu. ft.
Cut off of high pressure cylinder	=	$\frac{1}{4}$ stroke.
Cut off of low pressure cylinder	=	$\frac{1}{2}$ stroke.
Initial pressure (absolute)	=	120 lbs.
Back pressure (absolute)	=	3 lbs.

Clearance and compression are not considered. The stroke of the high pressure cylinder begins at A. Steam is cut off at $\frac{1}{4}$ stroke at C.

Since steam is cut off at $\frac{1}{4}$ stroke, the ratio of expansion in the high pressure cylinder is 4. Then the volume of steam at D is 4 times that at C and since $p\ v = p_1\ v_1$, the pressure at D is $\frac{120 \times 1}{4} = 30$ pounds. Steam is released at D and passes to the receiver. The pressure at release, L, of the low pressure cylinder is found from the equation $p\ v = p_1\ v_1$. The volume of steam in the high pressure cylinder at cut off, is $\frac{5}{4}$ cubic feet and the volume at L is 15 cubic feet; then the total ratio of expansion is $15 \div \frac{5}{4} = 12$. Then the pressure at L is $\frac{120 \times 1}{12} = 10$ pounds.

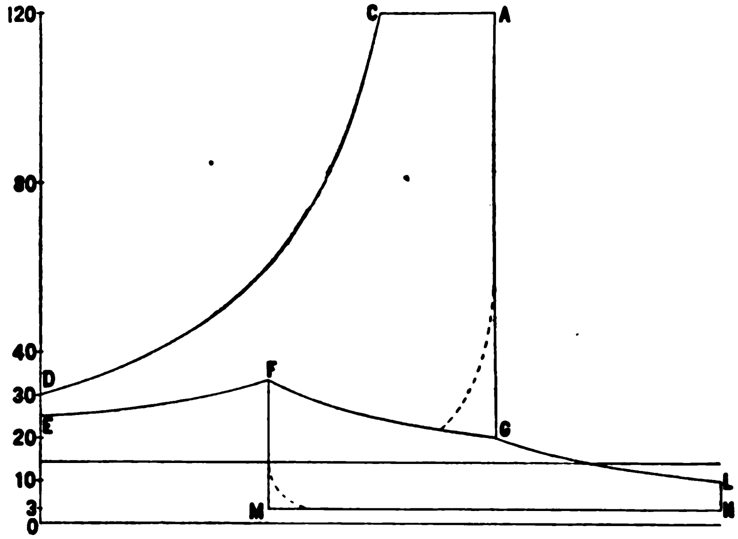

Fig. 28.

Since the low pressure cylinder cuts off at $\frac{1}{2}$ the stroke the volume at G is $\frac{1}{2}$ that at L.

Then as $p\ v = p_1\ v_1$, $p \times \frac{1}{2} = 10 \times 1$, or $p = 20$ pounds Hence the pressure at G is 20 pounds.

We can now find the pressure at E. The pressure in the receiver when the low pressure cylinder cut off, was 20 pounds because the low pressure cylinder was in communication with it.

Then as the piston in the small cylinder is at the end of the stroke when the steam is cut off in the large cylinder, steam at 30 pounds pressure rushes from the high pressure cylinder into the receiver and mixes with steam at 20 pounds pressure. When steam is cut off in the low pressure cylinder the high pressure cylinder and the receiver are in communication with each other; the total volume being the volume of the high pressure cylinder plus that of the receiver or $5 + 5 = 10$ cubic feet. What we wish to find is the pressure of the steam of this volume. The formula is,

$$P\,V = p\,v + p_1\,v_1.$$

Since there are 10 cubic feet in the receiver and high pressure cylinder, the value of V is 10. The volume of steam in the high pressure cylinder is 5 cubic feet and its pressure is 30 pounds. The volume of steam in the receiver is 5 cubic feet and its pressure is 20 pounds, then,

$$10\,P = 5 \times 30 + 5 \times 20$$

$$P = \frac{250}{10} = 25 \text{ pounds.}$$

Then the pressure at E = 25 pounds.

While the piston of the high pressure cylinder is on the return stroke, steam in that cylinder is compressed from E to F; the volume being 10 cubic feet. When the piston has completed $\frac{1}{2}$ the return stroke, the volume at F is equal to $\frac{1}{2}$ the volume of the high pressure plus the volume of the receiver or $\frac{5}{2} + 5 = 7\frac{1}{2}$ cubic feet.

Then with the formula $p\,v = p_1\,v_1$ we obtain,

$$p = \frac{25 \times 10}{7\frac{1}{2}} = 33.33$$

or the pressure at F is 33.33 pounds.

Then as the cranks are 90° apart, and the high pressure piston is at the middle of the stroke, the low pressure piston must be at the beginning of its stroke; or the pressure in the low cylinder is the same as that in the high and in the receiver, or 33.33 pounds. Since the stroke is beginning in the low pressure cylinder, steam is being admitted to it from the receiver and consequently the pressure in the large cylinder falls as the volume increases, which is shown by the line F G. Cut-off occurs in the low pressure cylinder at G, and the steam expands until the piston reaches L; the curve being an equilateral hyperbola. At L the release occurs and the pressure drops to that in the condenser; in this case about 3 pounds. The back pressure line is of course the pressure in the condenser.

The cards of Fig. 29 are from a cross compound engine. The rise of pressure in the middle of the back pressure line of the diagram from the small cylinder is due to the fluctuation of pressure in the receiver.

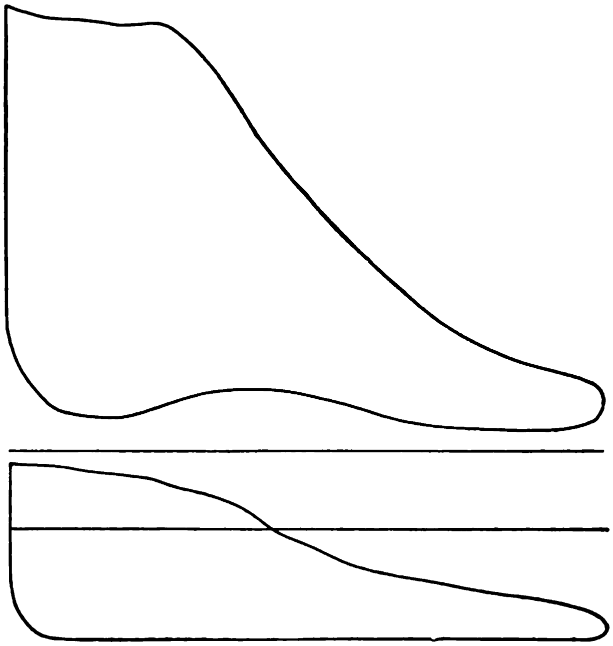

Fig. 29

COMBINED DIAGRAMS.

The indicator cards of multi-cylinder engines may be combined so that the pressures and volumes are shown in their proper relations. To do this, the cards are reduced to the same scale of pressure and the same scale of volume. To make the combined diagram of convenient size, the length of the low-pressure card is left as it is and the length of the high-pressure diagram is shortened in the ratio of the cylinder volumes.

Perhaps the best way to show the method is by an illustration. The cards shown in Fig. 30 were taken from a compound condensing engine.

Ratio of cylinder volumes = 1 : 3
Initial pressure = 138 pounds
Spring, high-pressure card = 60 pounds
Spring, low-pressure card = 30 pounds

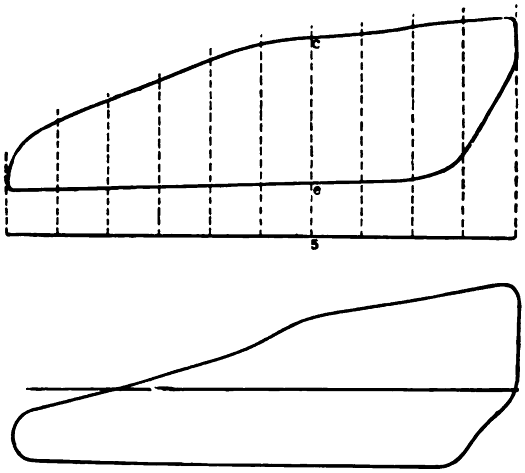

Fig. 30.

First draw the line of zero pressure A B and the line of zero volume E F (see Fig. 31). The line O P is drawn parallel to E F and at a distance proportional to the clearance of the low-pressure cylinder. Similarly the distance between M N and E F represents the clearance of the high-pressure cylinder. Now draw the atmospheric line C D. In this case it will be .49 inch above A B because a 30-pound spring is used, and $\frac{14.7}{30} = .49$. Reproduce the low-pressure card without change, as shown.

Divide the high-pressure card with 10 (or more) ordinates, and reproduce it with volumes and pressures of the same scale as the low-pressure card. Since a 60-pound spring was used, each ordinate will be twice as long (because $\frac{60}{30} = 2$). The distance between the ordinates will be $\frac{1}{3}$ as great because the high-pressure cylinder is $\frac{1}{3}$ the volume of the low. Draw the ordinates as shown. The distance between ordinates L and M will be $\frac{1}{3}$ of the length of the high-pressure card. The points *c* and *e* are on the fifth ordinate, and are twice as far above the atmospheric line as they are in the original high-pressure card.

After locating all the points draw the curve through them. Now draw the theoretical expansion curve R S through the point of cut-off of the high-pressure cylinder by the method explained on page 39. The difference between the area included between the theoretical curve and the lines of no pressure and no volume, and the sum of the actual areas, represents approximately the losses.

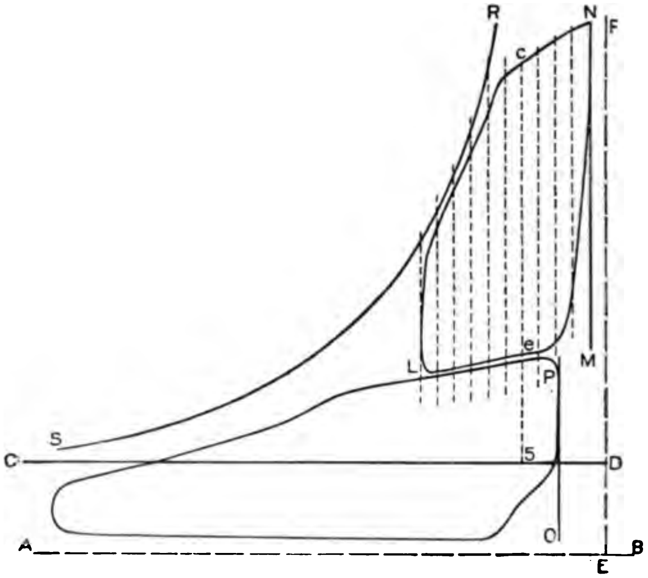

Fig. 31.

This method is not strictly accurate, because all the steam used in the high-pressure cylinder does not pass to the low-pressure cylinder; a small portion is left for compression. By thus combining the cards the action of the valves may be discussed, provided such data as size, type and speed are known.

The combined diagram of a multi-expansion engine drawn according to the above method is shown in Fig. 32. The volumes and pressures are first reduced to the scale of the low-pressure card. The cards are then redrawn at the proper distances from the lines of no pressure and no volume; the clearance in each cylinder being considered.

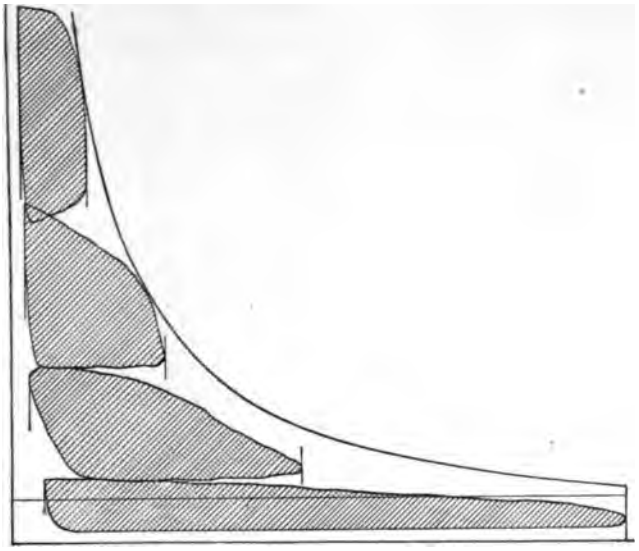

Fig. 32

HORSE-POWER OF COMPOUND ENGINES.

The I. H. P. of multi-cylinder engines may be found by adding the I. H. P. of the several cylinders. Another method is to reduce all the pressures to the area of the low-pressure cylinder. This is done by dividing the M. E. P. of each cylinder by the inverse ratio of its volume to that of the low-pressure cylinder.

Suppose the M. E. P. of the high-pressure cylinder of a compound engine is 78 pounds as found from the indicator card. If the volume of the high-pressure cylinder is $\frac{1}{3}$ that of the low, the M. E. P. of the high referred to the low would be $\frac{78}{3} = 26$ pounds. Then if the card from the large cylinder shows a M. E. P. of 30 pounds, the total M. E. P. is $30 + 26 = 56$ pounds, and the I. H. P. is found by inserting 56 as the value of P and using the area of the low pressure as A, in the formula for I. H. P.

INDICATOR CARDS.

The indicator reveals defects in the steam distribution. That is, if the valve or valves are set so that the events of the stroke are *too early* or *too late* or if *more work* is done at one end of the cylinder than at the other, the engineer finds it out by examining the indicator card.

Sometimes an engine appears to run well and the owner is perfectly satisfied with it; but from the indicator diagrams it is found that considerable saving might be effected by correcting the defects in the valve setting.

On looking at a diagram one might say it was a faulty card and yet for that type, size and speed it is perhaps the best that could be obtained from the engine. The same form of diagram is not possible from different types of engines. The diagram from a Corliss engine, having four valves, is different from that of the plain slide valve engine; also the diagrams from high speed engines differ from those of low speed.

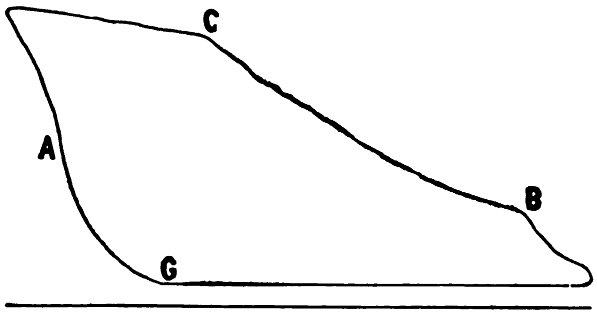

Fig. 33.

The most common faults in the distribution of steam in the cylinder can be divided into four classes.

a = **Admission** too early or too late.
b = **Cut-off** too early or too late.
c = **Release** too early or too late.
d = **Compression** too early or too late.

In the following figures,

A is the point of admission,
C is the point of cut off,
B is the point of release,
G is the point of compression.

The diagram shown in Fig. 33 shows **too early admission.** The admission line curves *backward* instead of being straight and perpendicular to the atmospheric line as it is in Fig. 24. The diagram also shows cut off, release and compression early. When the valve is of the plain slide valve type *all* the events are likely to be too early if one of them is. The reason for the event being too early is that the eccentric has too much *angular advance.* Hence the remedy is to decrease the angular advance.

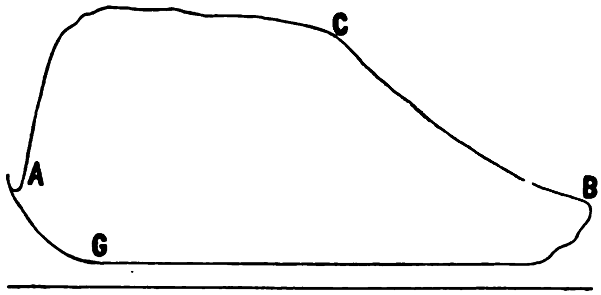

Fig. 34.

Fig. 34 shows a diagram having the opposite defects to those of Fig. 33. The events are too late. The admission line curves forward and the line shows that admission does not take place until after the stroke is well begun. Release occurs at the end of the stroke. In this case the angular advance of the eccentric should be increased until the admission line is perpendicular to the atmospheric line.

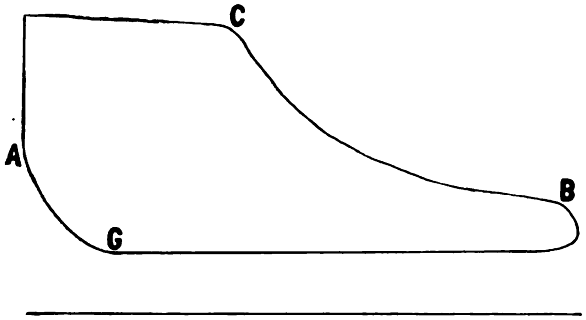

Fig. 35.

Fig. 35 shows a card having too much back pressure. This may be due to a small exhaust port or pipes, or the passage of steam through coils of pipe for heating. The card shows the other events to be good. A diagram having **too late cut off** is shown in Fig. 36. The pressure at release is high. When the engine is running under this condition much of the benefit from expansion is lost.

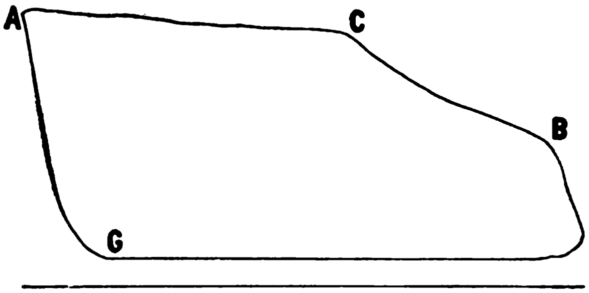

Fig. 36.

A diagram from a condensing engine is shown in Fig. 37. These oscillations are caused by the vibration of the indicator piston and spring. To avoid these vibrations never use a very light spring for high speed.

The diagram of Fig. 21 shows **too early cut off.** In this case the cut off is so early that the expansion line extends *below* the atmospheric line making a loop. The area of the loop must be *subtracted* from the area of the card as explained on page 33.

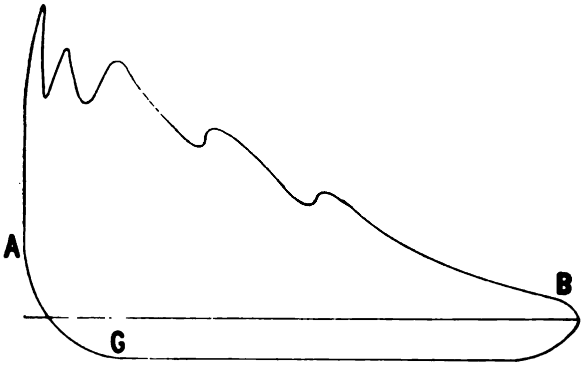

Fig. 37.

Fig. 38 shows a pair of diagrams from a plain slide valve engine. The admission lines are good. The sloping steam lines show wire-drawing due to the slow action of the valve or too small ports or pipes. This wire-drawing decreases the area of the card which indicates loss. The greatest fault is the inequality of area of the diagram for the ends. The late cut off and consequent late compression of one end causes more area than the too early cut off and too early compression of the other end. These cards can be improved by adjusting the angular advance of the eccentric and the length of the valve rod.

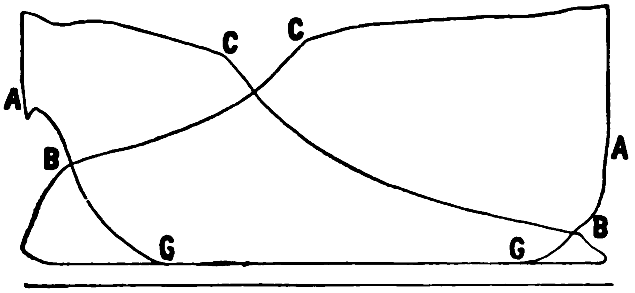

Fig. 38.

The diagram of Fig. 39 indicates too early compression. The compression curve extends above the initial pressure line. The area of the loop must be subtracted from the card area, when computing the I. H. P. If the cut off is kept the same and the compression made what it should be, the gain in area would be the area included between the full line and the dotted line plus the area of the loop. The remedy for this case is to decrease the inside lap.

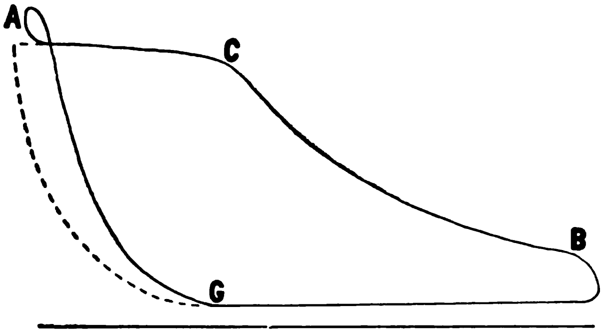

Fig. 39.

The amount of compression varies with the speed and type of engine. Slow speed engines require *less* compression or cushioning than high speed. The exhaust steam should *never* be compressed higher than the boiler pressure.

For high speed engines the compression should extend to about .9 the initial pressure. For medium speeds about .5 and for low speed .2 to .4.

In the case of a slide valve engine it is not always possible to set the valve so that the card may have all the events as they should be. Sometimes the laps of the valves should be altered. For too much compression decreases the inside lap. For too early cut off decrease the outside lap.

If the valve travel is increased, compression is retarded, that is, decreased; release occurs sooner.

STEAM CONSUMPTION.

The amount of steam used by an engine is called its **steam consumption** and for comparison, it is customary to state the amount of steam consumed per indicated horse-power per hour. By means of the indicator diagram the steam consumption can be computed approximately.

To find the Steam Consumption from the Diagram. The diagram shown in Fig. 40 is from a 20 × 36 engine, running at a speed of 80 revolutions per minute. A 40-pound spring was used.

By measuring the card, we find the mean ordinate to be .91 inch and the M. E. P. $= .91 \times 40 = 36.4$ pounds.

$$\text{I. H. P.} = \text{Engine Constant} \times \text{M. E. P.} \times \text{piston speed.}$$

$$= .00952 \times 36.4 \times 480.$$

$$= 166.33.$$

In Fig. 40 L M is the atmospheric line and O R the line of zero pressure drawn so that O L = 14.7 pounds. O N is the clearance volume = 8 per cent of the piston displacement. The line P Q is drawn from O R to some point on the compression curve. From D, a point on the expansion curve before release, the line D F is drawn perpendicular to O R.

Then from the diagram,

O R = 3.24 inches.
O F = 3.00 inches.
O P = .345 inch.
D F = .795 inch.
P Q = .795 inch.

The length of stroke is 36 inches or 3 feet, and the length of the diagram 3 inches. Then 1 inch of the length of the card corresponds to 1 foot of the stroke. The scale of spring used is 40. Therefore we can easily reduce the above dimensions to pounds pressure and to feet.

O R = 3.24 feet.
O F = 3.00 feet.
O P = .345 feet.
D F = 31.80 pounds.
P Q = 31.80 pounds.

The area of the piston (head end) is,

$$\frac{\pi\ d^2}{4} = \frac{3.1416 \times 400}{4} = 314.16 \text{ sq. in.} = 2.18166 \text{ sq. ft.}$$

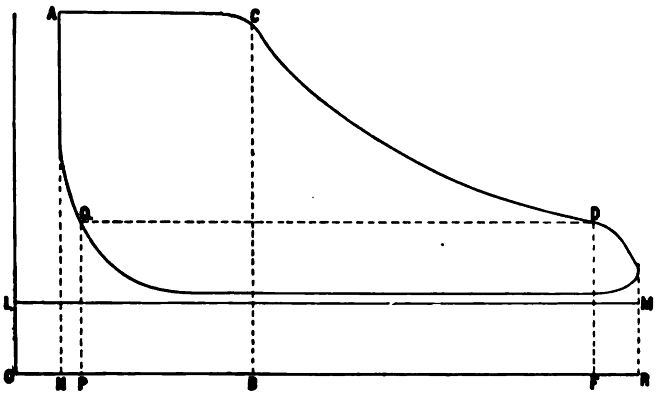

Fig. 40.

We can now find the volume of steam at any point of the stroke.

When the piston is at D, the volume is,

$$2.18166 \times 3 = 6.54498 \text{ cubic feet.}$$

When the piston is at Q, the volume is,

$$2.18166 \times .345 = .75267 \text{ cubic foot.}$$

From the steam tables we can find the weight of a cubic foot of steam at a given pressure.

The weight of 1 cubic foot at 31.8 pounds absolute pressure is .07773 pound. Then the weight of steam present when the piston is at D is,

$$6.54498 \times .07773 = .50887 \text{ pound.}$$

The weight of steam present when the piston is at Q is

$$.75267 \times .07773 = .05852 \text{ pound.}$$

The weight of steam in the cylinder is .50887 pound and the weight of steam kept for compression is .05852 pound. The weight exhausted per stroke is,

$$.50887 - .05852 = .45035 \text{ pound.}$$

The amount used per I. H. P. per hour is,

$$\frac{.45035 \times 2 \times 80 \times 60}{166.33} = 25.993 \text{ pounds.}$$

This may be stated in words as follows:

Measure the pressure from the vacuum line to some point in the expansion curve before release and from the steam tables find the weight of a cubic foot at that pressure. Multiply the volume (in cubic feet) of the cylinder, including clearance to that point, by the weight per cubic foot. The result is the weight of steam in the cylinder. As this weight includes the steam used for compression it must be corrected to obtain the weight used per stroke.

Take some point on the compression curve and measure its absolute pressure. Then compute the weight of steam to this point. Subtract this weight from the weight of steam to the point in the expansion curve and the result is the weight of steam used per stroke.

Multiply the weight of steam used per stroke by the number of strokes and divide by the indicated horse-power as found from the card. The final result is the number of pounds of steam consumed per horse-power per hour.

We may also calculate the steam consumption by taking the point of the expansion curve near the cut-off.

O B = 1.23 inches = 1.23 feet of the stroke.

B C = 1.8 inches = 72 pounds.

The weight of 1 cubic foot of steam at 72 pounds absolute pressure is .1671 pounds. Then the volume of steam at C is,

$$2.18166 \times 1.23 = 2.68344 \text{ cubic feet.}$$

The weight at C is,

$$2.68344 \times .1671 = .44840 \text{ pound.}$$

The weight kept for compression is the same as previously found, *i. e.*, .05852 pound.

Then the weight of steam used per stroke is,

$$.44840 - .05852 = .38988 \text{ pound.}$$

The steam consumption per I. H. P. per hour is,

$$\frac{.38988 \times 2 \times 80 \times 60}{166.33} = 22.50 \text{ pounds.}$$

If the valve doesn't leak, the amount of steam just after cut off should equal the amount just before release, but our calculation shows that there is $.45035 - .38988 = .06047$ pound more at release than at cut off. This shows that at entrance .06047 pound was condensed before the piston reached C and was re-evaporated before release. This calculation gives an idea of the amount of cylinder condensation.

If there is considerable compression as in Fig. 40 the above method may be simplified by taking the two points D and Q at the same height above the vacuum line. The pressures will then be the same.

Let W = weight of steam used per I. H. P. per hour.
w = weight of one cubic foot of steam at the absolute pressure D F.
L = length of the diagram, N R.
l = distance between Q and D.
P = mean effective pressure.

Then,

$$W = \frac{13{,}750 \times w \times l}{P\,L}.$$

Example. A card was taken from a 12 × 14 engine. Length of card L = 3.5 inches, l = 2.875 inches. Absolute pressure at

D = 30 pounds. M. E. P. = 30.75 pounds. **What is the steam consumption per I. H. P. per hour?**

$$\mathrm{W} = \frac{13{,}750\, w \times l}{\mathrm{P} \times \mathrm{L}}$$

$$= \frac{13.750 \times .0736 \times 2.875}{30.75 \times 3.5}$$

$$= 27.03 \text{ pounds.}$$

This method gives only an approximate steam consumption. On account of the leakage of valves and the initial condensation of steam in the cylinder, the actual consumption is somewhat greater. The excess varies considerably and makes all results so obtained of little value. In our calculation we used one diagram only, that for the head end; we assumed the diagram from the crank end to be the same. The indicated horse-power for each end should be computed, as it is subject to considerable variation. In the above formula the average mean effective pressure should be used. The results of calculations of steam consumption from indicator cards are so unreliable that engineers place little dependence upon them. The results may be used as a basis for estimates, but for accurate knowledge, an engine test must be resorted to

EXAMPLES FOR PRACTICE.

1. Given the following data to find the M. E. P. Area of card 2.79 square inches, length of card 3.1 inches, scale of spring 50. Ans. 45 pounds.

2. Steam enters a cylinder at a pressure of 210 pounds by gage and leaves at 3 pounds pressure (absolute). What is the thermal efficiency? Ans. 29 per cent.

3. An engine has a B. H. P. of 78.96. The I. H. P. is 86.73. What is the mechanical efficiency. Ans. 91 per cent.

4. An engine develops 149.97 I. H. P. If the piston diameter is $17\frac{3}{4}$ inches, the stroke 30 inches, and the M. E. P. 40 pounds, what is the speed? Ans. 100 revolutions per minute.

5. The theoretical M. E. P. of a triple expansion marine engine is 48 pounds. What is the probable actual M. E. P.?

6. The cylinders of a triple expansion are 12, 30, and 75 inches in diameter respectively. The stroke is 24 inches. The

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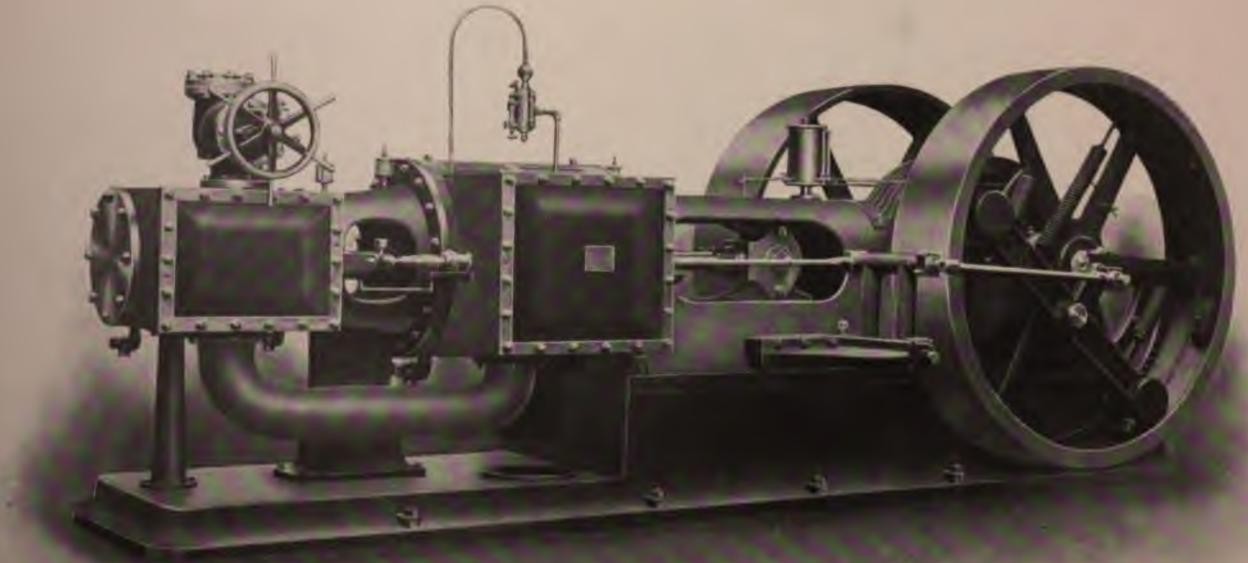
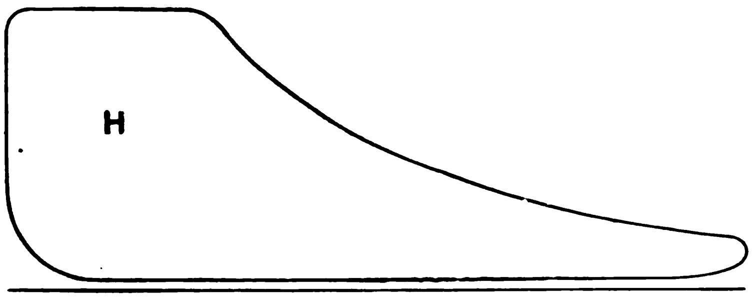


17"-28"-20" COMPOUND McEWEN ENGINE.
Ridgway Dynamo and Engine Co.

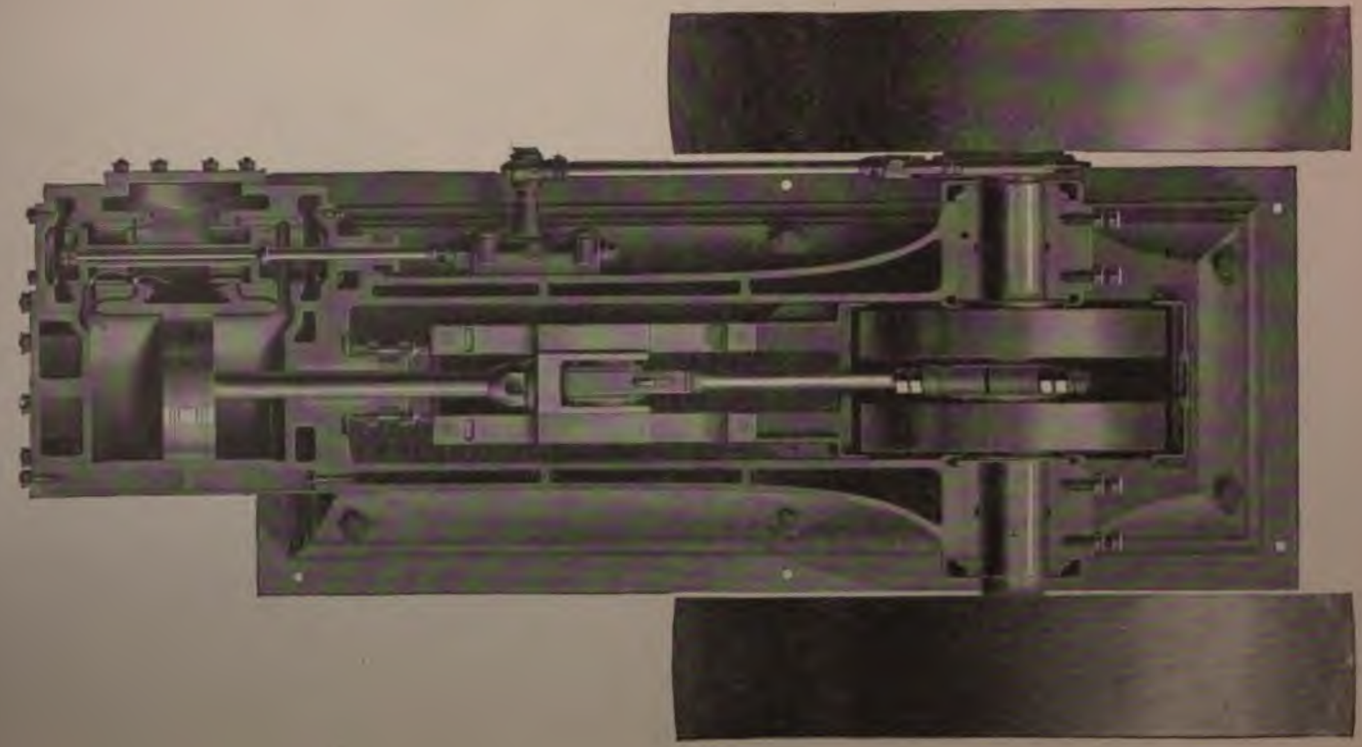
mean effective pressures from the cards were 118.4, 51.6 and 9.34 pounds respectively. What is the I. H. P. when the speed s 125 revolutions per minute? Ans. 2,050 I. H. P.

7. Find the steam consumption from the following card. The engine was running at 75 revolutions per minute, and developing 230 I. H. P. when the card was taken. A 60-pound spring was used, and the M. E. P. for this end was 46 pounds. Assume rank-end card to be the same as the head end.

Ans. 22 pounds (about).



Suggestion: First draw a line .3 inch above the atmospheric line, hen reduce this ordinate to *absolute* pressure, and use formula at he bottom of page 55.



HORIZONTAL SECTION OF ENGINE
New York Safety Steam Power Company.

VALVE GEARS.

Steam enters the cylinder of the engine through ports which must, in some manner, be opened and closed alternately, in order to admit and exhaust the steam at the proper time. To accomplish this purpose a valve is moved back and forth across the port openings. A complete understanding of the valve and valve gear is essential to the engineer as well as to the designer, for even though a valve be properly designed, its economy may be seriously impaired by improper setting. The design and adjustment of these valves plays a very important part in the efficient action of the steam engine.

The term "valve gear" includes the valve or valves that admit steam to and exhaust it from the cylinder of the engine, together with the mechanism from which the valves derive motion. There may be a single valve to regulate admission and exhaust, or there may be a double set of valves; one set to admit the steam at each end and another to release it. The valve may have a plain reciprocating motion, moved by a rod, or it may be opened by some device that lets go at the proper time, allowing the valve to drop shut under the influence of counter weights, springs or vacuum dashpots. To the first class belong the plain slide valve and its modification of piston valve, gridiron valve, etc.; to the second belong such valves as the Corliss, Brown, and others.

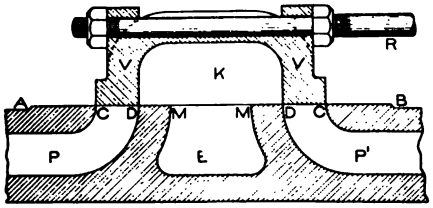


Fig. 1.

The simplest type of valve is the plain slide or D valve as shown in Fig. 1.

In this figure V is the valve, R the valve rod, K the exhaust cavity, P and P' the steam ports, E the exhaust port, AB the valve seat, and DM the bridges of the valve seat. The valve seat must be planed perfectly smooth, so that pressure on the valve will make a steam tight fit, and cause as little friction as possible when the valve slides. Furthermore, the length of the seat AB must be a little less than the distance from the extreme right-hand position of the right-hand edge of the valve to the extreme left-hand position of the left-hand edge of the valve. This allows the valve at each stroke slightly to over travel the seat, thus keeping it always worn perfectly flat and smooth. If the valve seat were not raised slightly above the rest of the casting, or if it were too short, the constant motion of the valve would soon wear a hollow path in the valve seat, and it would cease to be steam tight.

Eccentric. The valve usually receives its motion from an eccentric which is simply a disc, keyed to the shaft in such a manner that the center of the disc and the center of the shaft do not coincide. It is evident that as the shaft revolves, the center of this eccentric disc moves in a circle about the shaft as a center, just as if it were at the end of a crank. The action of the eccentric is equivalent to the action of a crank the length of which is equal to the eccentricity of the eccentric (the distance between the center of the eccentric and that of the shaft).

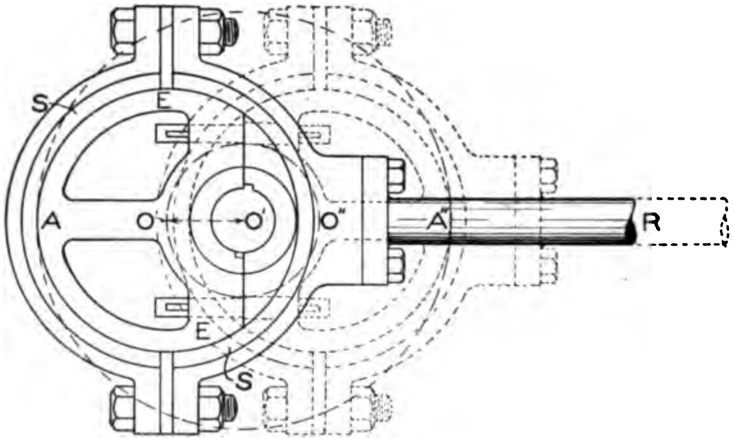


Fig. 2.

Fig. 2 represents the essentials of an ordinary eccentric. O is the center of the shaft, O the center of the eccentric disc E, and S is a collar encircling the eccentric and attached to the valve rod R.

As the eccentric turns in the strap, the point O moves in the dotted circle around O′, and the point A also moves in a circle. When half a revolution is accomplished the point O will be at O″, the point A will be at A″, and the eccentric strap and valve rod will be in the position indicated by the dotted lines.

Since the revolving shaft transmits motion to the valve through the eccentric, it will be necessary to study the relative motions of the crank and eccentric in order to get a clear idea of the steam distribution.

The distance of the center of the eccentric from the center of the shaft (OO′ in Fig. 2) is known as the eccentricity, or throw, of the eccentric. The travel of the valve is twice the eccentricity.

Valve without Lap. Fig. 3 shows a section through the steam and exhaust ports of an engine, together with a plain slide valve placed in mid-position, and so constructed that in this position it just covers the steam ports and no more. A valve is in mid-position when the center line of the valve coincides with the center line of the exhaust port.

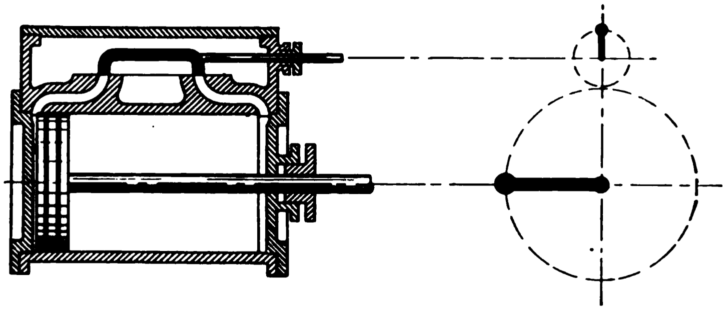


Fig. 3.

Fig. 1 shows the same valve drawn to a larger scale.

Suppose the valve is moved a slight distance to the right; the port P (see Fig. 1) is then uncovered and opened to the live steam which enters the cylinder and causes the piston to move. Since the two faces of the valve are just sufficient to cover the steam ports, it is evident that as the port P opens to live steam, the port P′ opens to the exhaust. The ports are closed only when the valve is in mid-position. This allows admission and exhaust to continue during the whole stroke. With such a valve there is no expansion or compression; the indicator card would be a rectangle, and the M. E. P. would be equal to the initial steam pressure, assuming no frictional losses in the steam pipe or condensation in the cylinder.

For a theoretical discussion of valve motion, it is assumed that the eccentric rod moves back and forth in a line parallel to the center line of the engine. This is not the case in practice, for the eccentric rod always makes a small angle with the center line, just as the connecting rod does, but the eccentricity is so small in comparison with the length of the eccentric rod that the angularity of the eccentric rod is very much smaller than the angularity of the connecting rod, and its influence may be neglected without appreciable error.

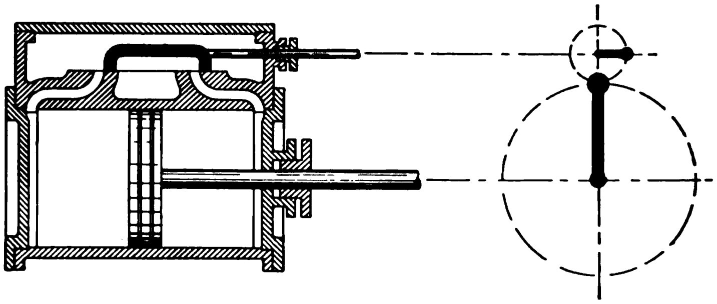


Fig. 4.

When the valve shown in Fig. 3 is in mid-position, the crank is on dead center, the eccentric is set at right angles to it, and the piston is just ready to begin the stroke.

Fig. 4 shows the relative positions of crank, piston, eccentric and valve when the crank has made a quarter turn or the piston has moved to half stroke. The eccentric is now in its extreme position to the right, the valve has its maximum displacement and both the steam and exhaust ports are wide open. The valve will not close again until the piston has reached the end of its stroke.

This type of valve is used only on small and unimportant engines, and since it allows no expansion of the steam, is very uneconomical. Furthermore, it will be seen that this valve opens just after the stroke begins, which is impractical, for it means that the piston has begun its stroke before the full steam pressure reaches it, which will cause an inclined admission line on the indicator diagram.

Valve with Lap. If the face of the valve is made longer than shown in Fig. 1, so that in mid-position it overlaps the steam ports, we shall have a valve such as shown in Fig. 5. The amount that the valve overlaps the steam ports is called the lap of the valve. In Fig. 5, DI is the *inside* lap, and OC the *outside* lap. It will at once be seen that both the admission and exhaust ports may remain closed during a part of the stroke, thus making expansion

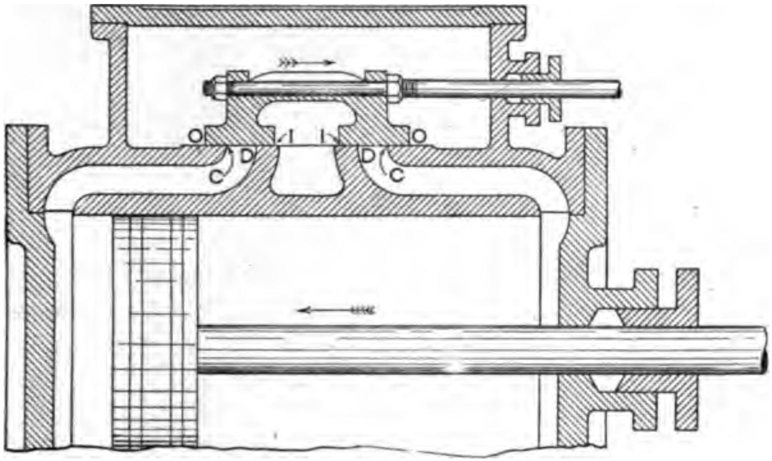


Fig. 5.

and compression possible. It is also evident that steam cannot be admitted until the valve uncovers the port by moving a distance from mid-position equal to OC. Admission continues until the valve returns to such a position that the outer edge of the valve again closes the port. Release will begin when the inner edge of the inside lap begins to uncover the port.

Fig. 6 represents a valve with lap, at the point of admission. Since the valve must move a distance equal to the outside lap before admission can take place, it is evident that the eccentric can no longer be at right angles to the crank at the beginning of the stroke, but must be ahead of the right-angle point by an amount equal to AOC. The angle AOC is known as the angular advance.

The maximum displacement of the valve is attained when the eccentric is horizontal as shown in Fig. 7. In this position both the steam and exhaust ports are wide open, and any further motion of the piston will cause the valve to move toward its mid-position.

Admission continues until the valve returns to the position shown in Fig. 8. Here the outside lap just closes the left-hand steam port, cut-off takes place, and the steam already in the cylinder begins to expand. As the valve continues to move toward the left, the left-hand inside lap begins to uncover the left-hand port and releases the steam at the position shown in Fig. 10.

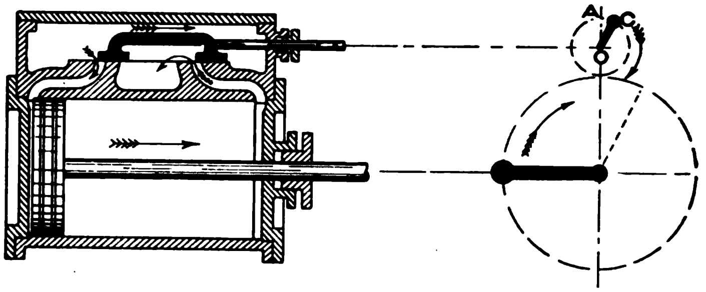


Fig. 6.

The dotted lines of Fig. 7 show the valve in its extreme position to the left. Any further movement of the piston will cause it to return toward mid-position.

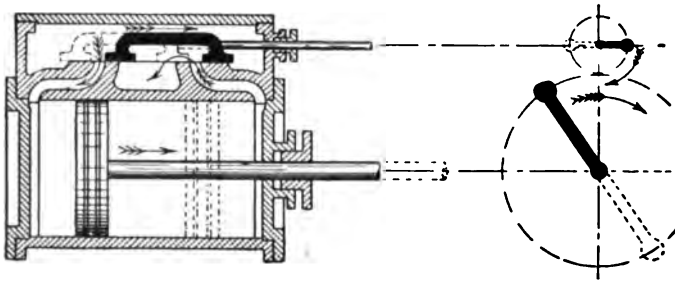


Fig. 7.

The dotted position of crank and eccentric in Fig. 10 shows the valve returned to the point of compression, which continues until the conditions of Fig. 6 are again reached and the opening valve allows steam again to enter the cylinder.

This process has been traced step by step for one end only; let us now consider what is happening at the other end.

Admission is the point at which the valve opens to admit steam to the cylinder. **Cut-off** is the point at which the valve closes to cut off the admission of steam. **Release** is the point at which the exhaust is opened; and **Compression** is the point at which the exhaust is closed.

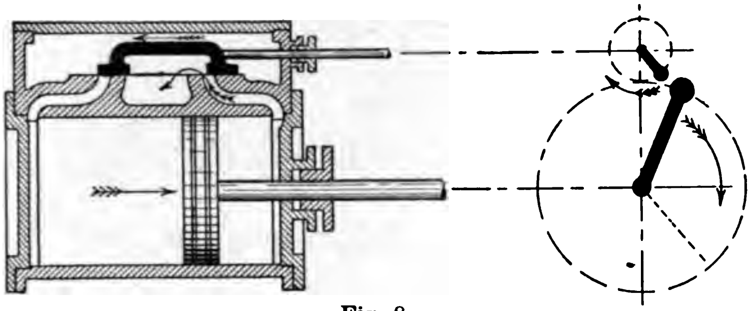


Fig. 8.

While the crank is moving from the position shown in Fig. 6 to that of Fig. 8, steam is being admitted to the head end and being exhausted from the crank end. The inside lap being less than the outside lap, causes the exhaust to continue longer than the admission.

Fig. 9 shows the relative positions of crank, eccentric and valve when the exhaust closes on the crank end and compression begins. Between these two positions the steam is expanding in the head end and exhausting from the crank end.

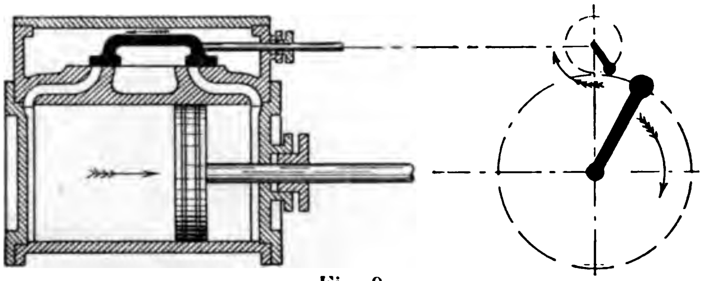


Fig. 9.

Between the positions of Fig. 9 and Fig. 10 both ports are entirely closed, expansion is taking place in the head end and compression in the crank end. Fig. 10 is head-end release. Fig. 11 shows admission at crank end of cylinder and marks the end of crank-end compression.

By referring to Figs. 6–11, the effect of any change of lap may at once be observed. If the outside lap is *increased*, the valve must move farther from mid-position before admission will occur, and on the return, after the maximum displacement is reached, the outside lap, being wider, will close the port sooner, and the cut-off shown in Fig. 8 will take place before the crank reaches the angle there shown. A decrease of outside lap will make cut-off later and admission earlier.

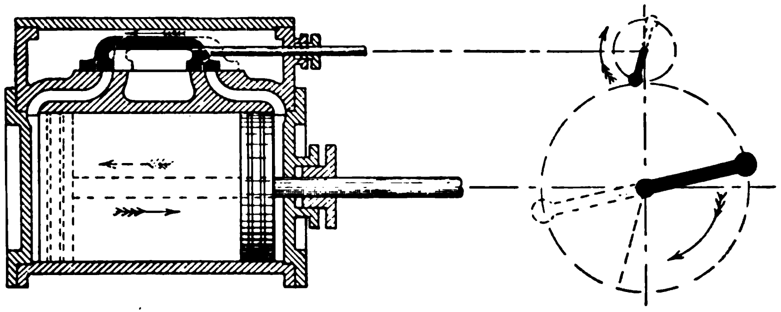


Fig. 10.

If the inside lap is increased, the valve must move farther before release occurs and the crank angle would be greater than shown in Fig. 10. On the return of the valve to the dotted position shown in Fig. 10, the port will close earlier and make an earlier compression; the crank angle will be less than is there shown. Decreasing inside lap will cause earlier release and later compression.

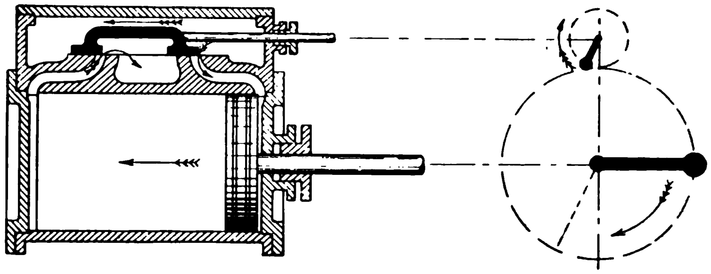


Fig. 11.

Thus we see that it is the outside lap that influences admission and cut-off, and the inside lap that controls release and compression. For this reason the outside lap is often called the *steam lap*, and the inside lap the *exhaust lap*.

Lead. If a valve having lap is in mid-position, the port is closed and the engine cannot start because no steam can enter the cylinder. That the steam may be ready to enter the cylinder at the beginning of the stroke it is necessary that the eccentric be set more than 90° ahead of the crank and the eccentric radius will take an angle as shown in Fig. 6, called the *angular advance.* In order that the ports and clearance may be properly filled with steam at the beginning of the stroke, it is necessary that the valve be displaced from its mid-position an amount slightly greater than the outside lap. With the piston at the end of the stroke the valve will have a position as shown in Fig. 12. The port will be open the distance AB. This causes the eccentric to be moved forward a slight amount in excess of the angular advance. This excess is called the *angle of lead.*

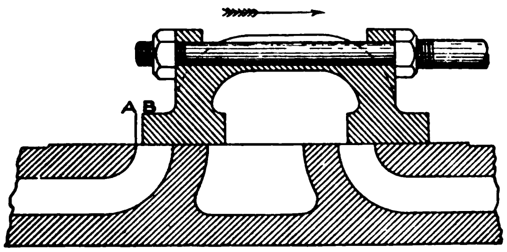


Fig. 12.

In Fig. 13, O'R' represents the crank at the beginning of the stroke, LOA the angular advance, and AOA' the angle of lead. The eccentric, to give lead, must be set at the angle ROA' ahead of the crank or 90° plus angular advance plus angle of lead. In large, quick-running engines, a liberal lead is essential, so that the ports and clearance may be well filled with steam before the stroke begins. If there is no lead, a portion of the steam will be used in filling these places and full pressure steam will not reach the piston until it is well advanced on the stroke. This will give a sloping admission line as shown

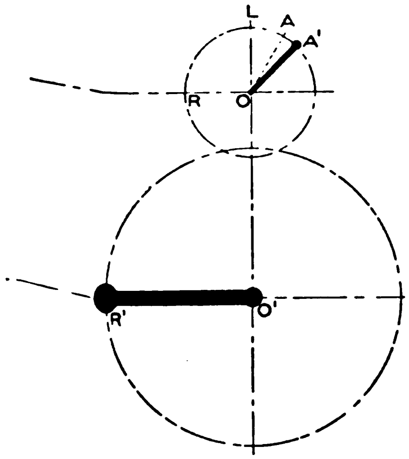


Fig. 13.

in Fig. 14. Too much lead, on the other hand, will cause too early an admission as shown in Fig 15.

If the angular advance is increased, the eccentric will be moved farther ahead of the crank, and consequently will begin its motion sooner. It will necessarily arrive at each of the events sooner than before. If then, the angular advance is increased, all of the events of the stroke will occur earlier.

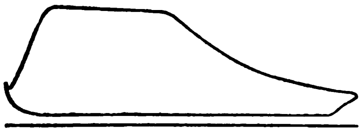


Fig 14.

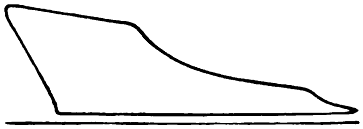


Fig. 15.

Inequality of Steam Distribution. In the valve diagrams thus far considered, the events of the stroke have been discussed for each end separately, without reference to the relation of similar events on the other side of the piston. If the connecting rod were of infinite length, so that it would always remain parallel to the center line of the engine, the distribution would be the same for both ends of the cylinder. In practice, the connecting rod is from 4 to 8 times the length of the crank, which causes the connecting rod always to be at an angle to the center line of the engine, and for a given crank angle makes the piston displacement greater at the head end than at the crank end.

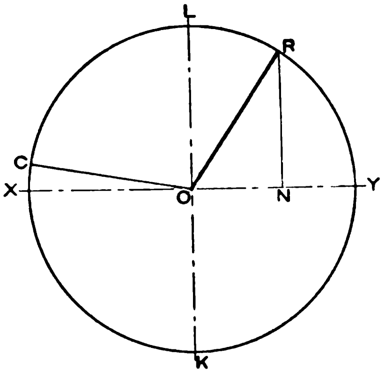


Fig. 16.

To find the displacement of the valve, let us consider Fig. 16. The circle represents the path of the eccentric center during a complete revolution of the engine. OC represents the crank, and OR the corresponding position of the eccentric. The diameter XY represents the extent of the valve travel. Since the eccentric rod is so long in comparison to the eccentricity, we make no appreciable error by assuming it always to be parallel to the center line of the engine. When the eccentric is at OL, the valve is in mid-position. At OR the valve has moved from mid-position an amount ON, found by dropping a perpendicular from R to the center line XY. If the angularity of the connecting rod could be neglected, the *piston* displacement could be found in the same manner.

To find the displacement of the piston, a diagram as shown in Fig. 17 must be drawn. In this figure AB represents the cylinder, P the piston, H the crosshead, HR the connecting rod, and OR the crank. Suppose now the engine should stop in this position and then be clamped. The piston displacement would be represented by AP. If the crank pin at R should now be loosened so as to allow the connecting rod to fall to a horizontal position, the point R would describe the arc of a circle RN, and XN would represent the piston displacement and would be equal to AP

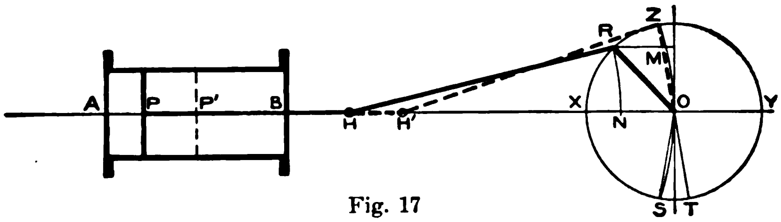


Fig. 17

Suppose now that in this disconnected way the piston, crosshead and connecting rod were moved forward until the end of the rod came to O. P would then be at P' and the piston would be in the middle of its stroke. Now swing the end of the rod up to its proper position on the crank-pin circle, the piston remaining stationary. It would describe an arc OZ. The crank pin would be at Z, less than a quarter revolution from X, while the piston would be in the middle of its stroke.

Suppose this engine were running with cut-off at half stroke on the head end and that XOZ represented the corresponding crank angle. On the return stroke the valve would cut off at the same crank angle YOT = XOZ, and OT would represent the crank cut-off on the return or crank-end stroke. The piston, as we have just seen, will not be at half stroke except when the crank is at OZ or OS. Consequently OT is less than half stroke and cut-off takes place earlier at the crank end than at the head end. When the crank is at OZ the eccentric will be at OA (Fig 17*a*), and the valve displacement will be OB. When the crank is at OT the eccentric will be at OA′, and the valve displacement will be OB′, which is equal to OB, the displacement of the valve at cut-off in the head end. The piston displacement will be OX in the head end and WY in the crank end when cut-off occurs. If the connecting rod always remained parallel to the center line, the cut-off would be the same at both ends.

Compensation of Cut-off. It has already been pointed out that lengthening the outside lap makes the cut-off earlier, and shortening the lap makes it later. The cut-off in the case just cited may then be equalized by altering the outside laps. If we increase the outside lap on the head end, or decrease the crank-end lap, the inequality will be less. By changing either or both of the laps the proper amount, the cut-off may be exactly equalized.

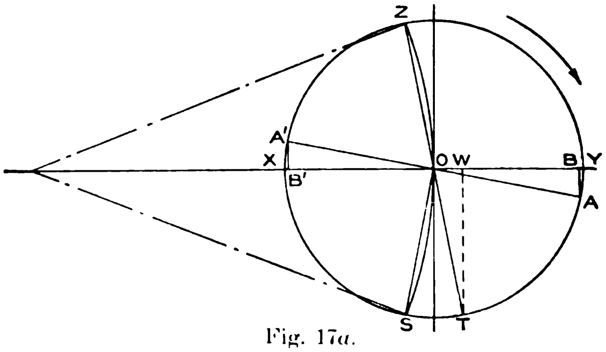


Fig. 17a.

But altering the outside lap changes the lead as has already been explained. If the lap is increased on the head end, the lead will be less than on the crank end. If the lead becomes too small on the head end, the angular advance may be increased but the inequality of lead will still remain, for this increase of angular advance will increase the lead at the crank end as well as at the head end, and by hastening all the events of the stroke may give a bad steam distribution if care is not taken.

Unequal lead is of less consequence on a low-speed than on a high-speed engine. On low-speed engines the cut-off may be equalized at the expense of lead with beneficial results, but on high-speed engines it will not do to give too little lead at one end. A high-speed engine requires more lead than a low-speed, for there is relatively less time in each stroke for the clearance to fill with steam.

If both inside laps are equal, compression will not occur equally at both ends. To equalize it, the inside laps may be changed in the same manner as the outside laps are changed to equalize the cut-off. By altering these inside laps to equalize compression, it may happen that the lap is reduced enough to leave the exhaust port open when the valve is in mid-position.

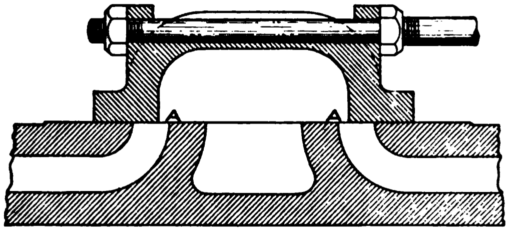


Fig. 18.

This opening of the valve is called an inside clearance, or *negative lap*. In Fig. 18, A is the inside clearance.

Rocker. Sometimes it happens that the valve stem and eccentric rod cannot be so placed that they will be in the same straight line; or it may be that the travel of the valve must be so great as to require an excessively large eccentric. In such cases a rocker may be used.

Fig. 19 shows a valve that is not in line with the eccentric. This occurs in horizontal engines when the valve is set on top of the cylinder instead of on one side. By means of the rocker AG the valve may receive its proper motion.

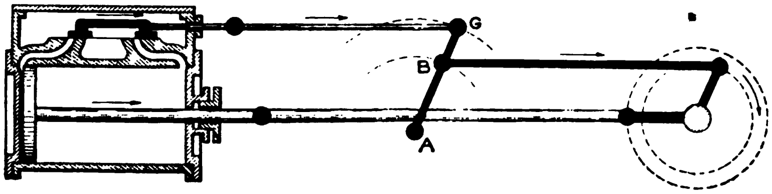


Fig. 19.

In case it is more convenient to place the pivot of the rocker arm between the connections to the valve stem and those of the eccentric rod, such an arrangement as shown in Fig. 20 may be used. Here it will be noticed that the valve stem and eccentric rod are moving in opposite directions, and to give the valve the same motion as in Fig. 19, the eccentric must be moved 180° ahead of the position there shown.

If AB is less than AG, the valve travel will be greater than twice the eccentricity, in proportion as AG is greater than AB. In all cases the valve travel is to twice the eccentricity as AG is to AB. Thus, if the valve travel is $4\frac{1}{2}$ inches, AB, 15 inches, and AG, 18 inches, then $\frac{15}{18} \times 4\frac{1}{2} = 3\frac{3}{4}$ inches, will equal twice the eccentricity.

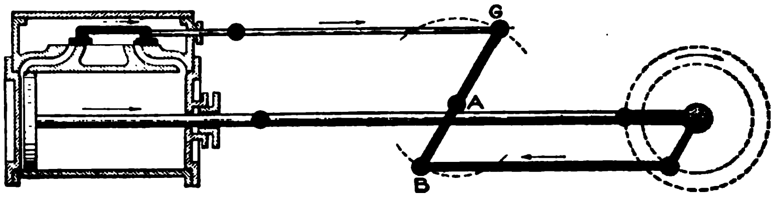


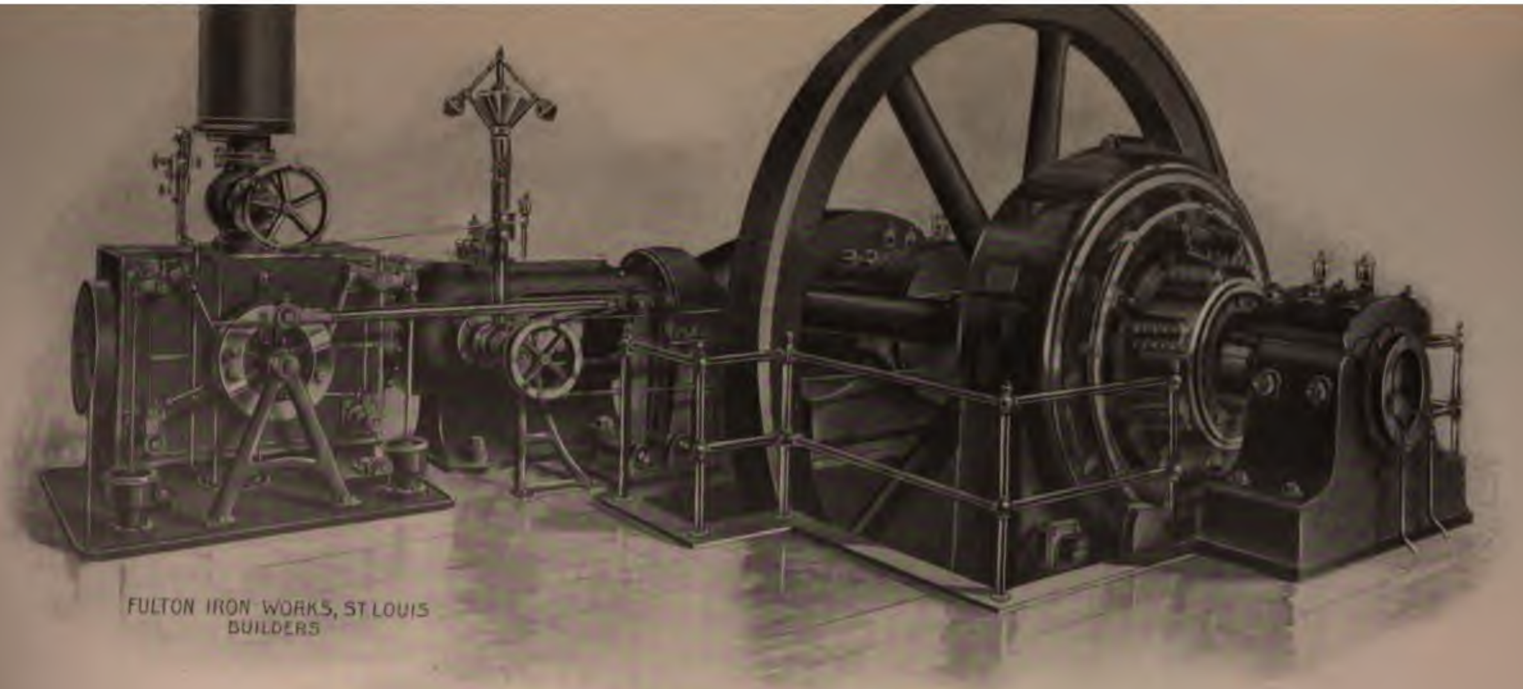
Fig. 20.

A valve gear may be so laid out as to make both the cut-off and the lead equal for both ends of the cylinder. This may be done by a proper porportion between the rocker arms, and a careful location of the pivot of the rocker. The eccentric must then be set accordingly. In this manner the Straight Line engine equalizes the cut-off and lead. A discussion of this method will be considered later.

VALVE DIAGRAMS.

Zeuner's Diagram. In order to study the movements of valves, the effect of lap, lead, eccentricity, etc., diagrams of various sorts have been devised. By the use of diagrams we may acquire a knowledge of valve motion without the complex mathematical expressions that such a discussion would entail. The most useful of these various diagrams is that devised by Zeuner, and to avoid complexity we shall confine ourselves to a discussion of this diagram alone. The eccentric rod is assumed to be of infinite length, and the positions of the crank are shown on the diagrams. The displacement of the piston can easily be found if the ratio of crank to connecting rod is known.

In Fig. 21 let OY be the eccentricity, then XOY will represent the valve travel, and the center of the eccentric will move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in the circle XWY. Let OR represent the position of the crank and Or the corresponding position of the eccentric, which is 90° + angle of advance θ ahead of the crank. Draw OW perpendicular to XY and lay off from it the angle WOM = angle of advance θ towards the crank. With OM as a diameter, construct a circle. OM is equal to the eccentricity, and the circle MPO is known as the *valve circle*. If OR, the center line of the crank, cuts this valve circle at P, then OP is equal to the displacement of the valve from mid-position.

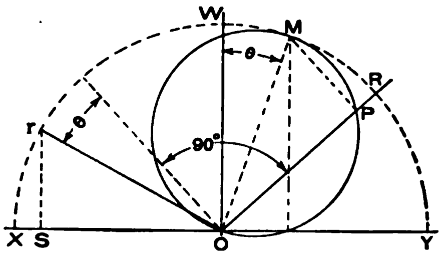

Fig. 21.

To prove this, draw rS perpendicular to XY. Since Or is the position of the eccentric, OS will represent the valve displacement from mid-position. Draw MP. Then by geometry OPM is a right angle because it is inscribed in a semicircle. OSr is also a right angle ; the two right-angled triangles OSr and OMP are equal because they are similar and have two corresponding sides equal. Or = OM, being radii of the same circle. But we have seen that OS is equal to the valve displacement, therefore OP is also equal to the valve displacement, for it is equal to OS.

Now that the truth of our proposition has been proved, let us see how we may study the valve motion from such a diagram. See Fig. 22. As before, let XY represent the valve travel, then the circle XEYF will represent the path of the center of the eccentric. Let θ be the angular advance and lay off EO toward the crank, making an angle θ with the vertical. Produce EO to F, and on OE and OF as diameters draw the valve circles as shown. Let the outside lap be an amount equal to OV, then with O as a center and OV as a radius draw an arc intersecting the upper valve circle at V and K. Lay off OP equal to the inside lap and with O as center and OP as a radius draw an arc intersecting the valve circle at P and Q. Draw the crank line AO passing through V. Then, when the crank is in this position, the displacement of the valve is equal to OV (the outside lap) and the steam is ready to enter the cylinder. This is the position of the crank at admission, and the crank angle XOA is called the lead angle. The valve has lead, therefore the admission takes place before the end of the stroke. When the crank reaches the position OE, the displacement of the valve is equal to OE, the eccentricity, and is the maximum displacement. Further motion of the piston causes the valve to move toward mid-position until, at the crank position OC, the displacement OK is again equal to the outside lap and the valve has reached the point of cut-off. When the position OH is reached, the crank line is tangent to both valve circles and there is no displacement of the valve. At this point the valve is in mid-position.

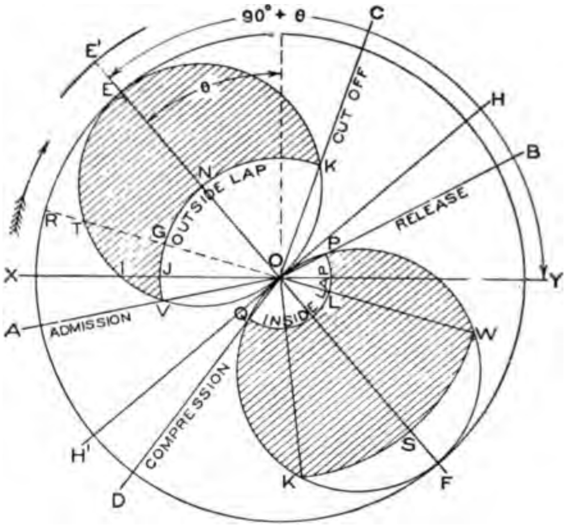

Fig. 22.

Further crank movement draws the inside lap toward the edge of the exhaust port until, at the crank position OB, the displacement is equal to OP (the inside lap) and release begins. At OF the maximum valve displacement is again reached and the valve moves in the opposite direction until at OD its displacement from mid-position, is again equal to OQ = OP = the inside lap, and compression takes place. At OH′ the valve is again in mid-position. At OX the displacement of the valve is OI, but since the valve has to move the distance OJ before the port begins to open IJ must represent the port opening when the crank is on dead center and by definition we know that lead is the amount of port opening at this position. Therefore IJ represents the lead.

At the position R, the port is open an amount equal to TG, at E the opening is a maximum equal to EN. At C the opening is nothing. If LW represents the total width of the steam port, the exhaust port will be open wide when the displacement of the valve is equal to OW and it will remain wide open while the crank swings from OW to OK.

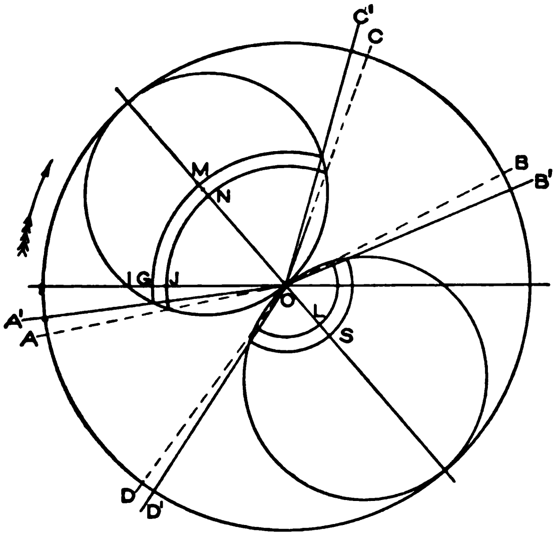

Fig. 23.

If the width of steam port in addition to the outside lap were laid off on the other valve circle it would fall at E′. For the admission port to be wide open, the displacement of the valve would have to be equal to OE′, which is more than the maximum displacement. This shows that in this case the steam port is never fully open and that the left-hand edge of the valve overlaps the right-hand edge of the port by an amount equal to EE′ when the valve has reached its maximum displacement.

Fig. 22, with its two valve circles, shows the diagram for the head end of the cylinder only. The crank-end diagram would be similar except that the laps might not be equal to those of the head end.

We are now in a position to consider more in detail the effect of changing in any way either the valve or the setting. Let us consider Fig. 23, which is in every way like Fig. 22 except that all unnecessary letters and lines are omitted to avoid confusion. If the outside lap is increased an amount equal to NM, the admission will take place later, at crank position OA′; the lead will be reduced to IG and cut-off will take place earlier at OC′. If the outside lap is *reduced* a like amount the contrary effects will be observed. If the inside lap is increased an amount equal to LS, the release will take place later at the crank position OB′ and compression will take place earlier at OD′. The contrary effect will be observed by decreasing the inside lap.

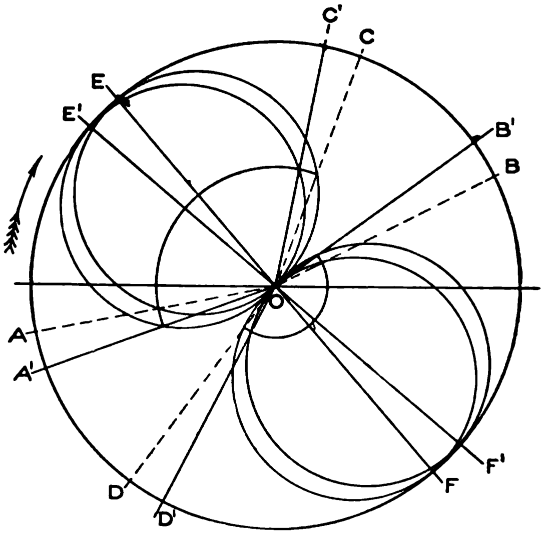

Fig. 24.

If the angular advance is increased (see Fig. 24) all the events will occur earlier. This is evident from the figure; the crank revolves in the direction indicated by the arrow and OA′ (new position of admission) is *ahead* of OA, the old position.

If the eccentricity is increased, Fig. 25, the valve travel will increase and admission will take place earlier at OA′; the lead will be increased an amount equal to II′, and cut-off will take place later at OC′. Release will be earlier at OB′ and compression will be later at OD′. The upper valve circle will now cut the arc drawn from O as a center, with a radius equal to the outside lap plus the width of steam port, in the points W′ and H′, and the admission port will be open wide while the crank is moving from OW′ to OH′. Similarly, the lower valve circle cuts the arc drawn from O as a center, with a radius equal to the inside lap plus the width of steam port, in the points W and H. The steam port is then wide open to exhaust while the crank is moving from W to H. From the above it will be seen that the periods are all changed by changing the travel; thus, admission and exhaust begin sooner and last longer, while expansion and compression begin later and cease sooner. With change in the angular advance, however (see Fig. 24), the periods are neither increased nor decreased.

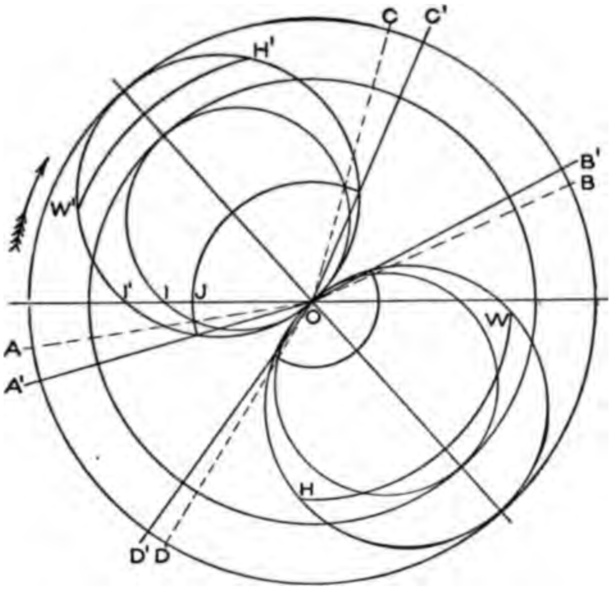

Fig. 25.

For convenience, these results are collected in the following table which shows the effect of changing the laps, travel, and angular advance:

	Increasing Outside Lap.	Increasing Inside Lap.	Increasing Travel.	Increasing Angular Advance.
Admission.	Is later. Ceases sooner.	Not changed.	Begins earlier. Continues longer.	Begins earlier. Same period.
Expansion.	Is earlier. Continues longer.	Beginning unchanged. Continues longer.	Begins later. Ceases sooner.	Begins earlier. Same period.
Exhaust.	Unchanged.	Occurs later. Ceases sooner.	Begins earlier. Ceases later.	Begins earlier. Same period.
Compression	Begins at same point.	Begins sooner. Continues longer.	Begins later. Ceases sooner.	Begins earlier. Same period.

PROBLEMS.

All the problems on valve gears involve the relations between certain variables which are:

The valve travel.
Angle of lead.
Outside lap.
Inside lap.
Points of stroke at which admission cut-off, release and compression take place.

In designing a Slide Valve, a few of these variables depend upon the conditions under which the engine is to run. For instance, the valve travel is limited, cut-off must be at a certain point and the engine must have a certain lead. Then, with the aid of a Zeuner's diagram, the remaining proportions of the valve may be determined.

Let us consider a few examples:

Given the valve travel = 3 inches.
Inside lap = ¾ inch.
Angular advance = 35°
Angle at cut-off = 115

To determine the laps, the lead and the crank angles at admission, compression and release.

In Fig. 26, let XY represent the valve travel = 3 inches. Draw OM perpendicular to XY, and on XY as a diameter draw the circle XMYF representing the path of the center of the eccentric as it revolves about the shaft. Lay off the angle MOE = the angular advance = 35° so that the angle XOE is equal to 90° *minus* the angular advance. Produce EO to F. Then on OE and OF as diameters draw the valve circles. The eccentricity OE or OF, if no rocker is used, will be half the valve travel. Lay off the crank angle XOC = angle of crank at cut-off = 115°, and OK will then represent the distance of the valve from mid-position when cut-off takes place. This distance we know is the outside lap. Draw the arc KI, known as the lap circle, and it will cut the valve circle again at V. When the valve is again the distance OV = the outside lap from mid-position, admission will take place. Draw the line OVA and this will represent the position of the crank at admission.

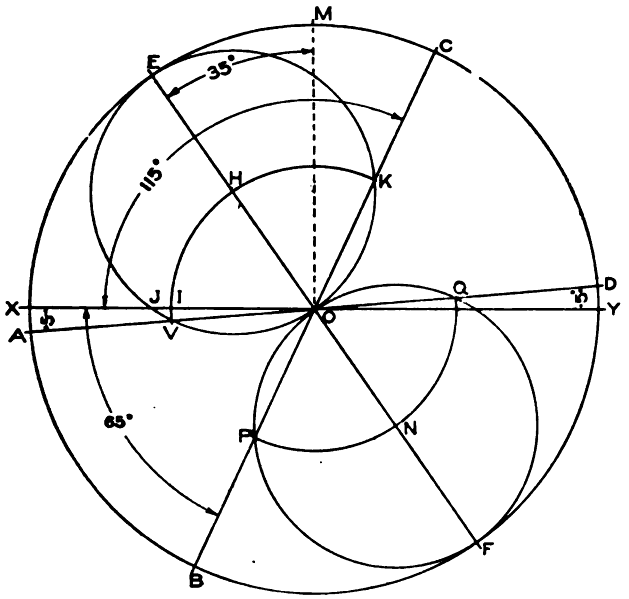

Fig. 26.

When the crank is at OX, the valve displacement is equal to OJ. This is at dead center and the valve is open the amount IJ, for it has moved this distance more than the outside lap. Therefore IJ is the lead for this end.

Now on the other valve circle, draw the arc PQ with the inside lap ($\frac{3}{4}$ inch) as a radius. It will cut the valve circle at P and Q. When the valve displacement is equal to OQ, the exhaust port has just closed and the engine is at compression. In the same way OP is the valve displacement at release when the port begins to open. OQD represents the crank position at compression and OPB the crank position at release.

The results then are as follows:

Given:
Valve travel = XY = 3 inches.
Angular advance = angle MOE = 35°.
Inside lap = OP = ¾ inch.
Crank angle at cut-off = angle XOC 115°

Found:
Outside lap = OK = ¾ inch.
Angle of lead = XOA = 5°.
Linear lead = IJ = $\frac{3}{32}$ inch.
Max. port opening for admission = HE = ¾ inch.
Crank angle at compression = XOD = 185°
Crank angle at release = XOB = 65°
Max. port opening for exhaust = FN = ¾ inch.

Fig. 26 is drawn full size, and all of these measurements may readily be verified. This figure is drawn for the head end only. If the crank angle at cut-off is the same on both ends, the Zeuner's diagram for the crank end will be exactly like Fig. 26.

ANOTHER PROBLEM.

Given:
The valve travel = 3 inches.
The lead angle = 6°.
Crank angle at cut-off = 70°.
Crank angle at compression = 75°.

To Find:
Angular advance.
Laps.
Linear lead.
Crank angle at release.

As before, let XY represent the valve travel = 3 inches and draw OM and the circle XMYF. See Fig. 27. Lay off the lead angle XOA = 6°. Then OA represents the crank position at admission. Next lay off the crank angle XOC, the angle at cut-off 70°. Bisect the angle COA by the line OE and on OE draw the valve circle. Angle MOE = the angular advance. The valve circle will cut the crank lines OC and OA at K and V respectively. If the work has been carefully done, OK will be exactly equal to OV and will represent the outside lap. The lead is IJ as before. Draw OD at position of the crank at compression so that angle XOD = 75°. Continue OE to cut the eccentric circle at F. On

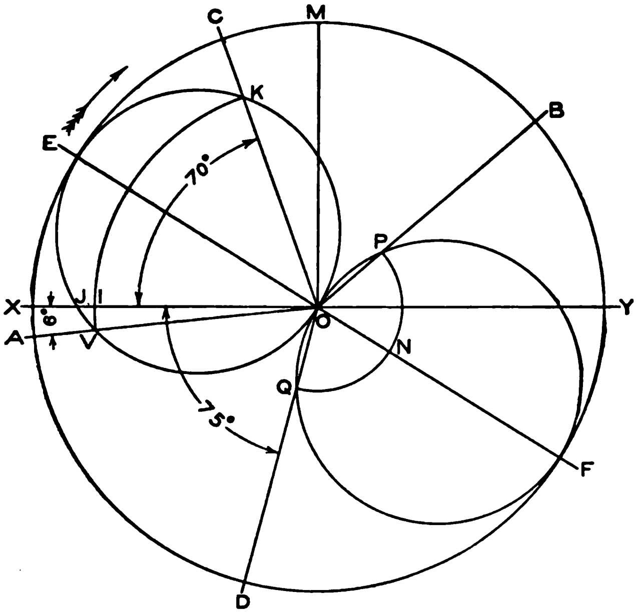

Fig. 27.

OF draw the second valve circle. It will cut OD at Q, and OQ will represent the inside lap. Draw the lap circle OP, and the crank position OPB. This will be the crank position at release.

The angular advance in this problem is large and all the events of the stroke are early. Compression and release are excessively early and the outside lap is unusually large. In the previous problem, with cut-off at about two-thirds stroke, the results were nearly normal. Cut-off with the plain slide valve, earlier than half stroke cannot be had without sacrificing the steam distribution on the other events.

To sum up we have

Given:

Valve travel = XY = 3 inches.
Lead angle = XOA = 6°.
Crank angle at cut-off = XOC = 70°.
Crank angle at compression = XOD = 75°.

To find:

Angular advance = MOE = 58°.
Outside lap = OV = 1 $\frac{6}{32}$ inches.
Lead = IJ = $\frac{3}{32}$ inch.
Inside lap = OQ = $\frac{7}{32}$ inch.
Crank angle at release = XOB = 130°.

Suppose in this last problem the cut-off had been given at half stroke instead of having the crank angle given, and that the compression had been given in the same way. We should, of course, need to know the ratio of length of connecting rod to crank. Let this be given as 4, that is, the connecting rod is four times the length of the crank.

In Fig. 28 let XY represent the valve travel. Extend XY to the left to the point Z, and make OZ equal to four times OX. With Z as a center and OZ as a radius, strike an arc OC that will cut the eccentric circle at C; then draw OC, which will represent the crank when the piston is at half-stroke, which is assumed to be the point of cut-off.

To find the crank angle at compression, lay off YH equal to .8 of the distance YX. From H lay off HW = OZ = four times OX. From W as a center with a radius WH, draw an arc cutting the eccentric circle at D. Draw OD, which will represent the position of the crank at compression.

The student is advised to read over again pages 13 to 14 if this explanation of finding the crank angle does not seem perfectly clear.

ANOTHER PROBLEM.

Given:

Cut off at .6 stroke.
Lead = $\frac{1}{16}$ inch.
Maximum port opening = $\frac{1}{2}$ inch.
Ratio of crank to connecting rod = 4.

To find:

The eccentricity.
Lead angle.
Angular advance.
Laps.

In Fig. 29 assume an eccentricity that will if possible be a little too large. Let us take for trial 2⅛ inches and draw XY equal to twice the assumed eccentricity equal to 4¼ inches. Lay off XC′ equal to .6 of XY, and with a radius equal to four times OX draw the arc C′C as already explained.

Then draw OC, which will represent the position of the crank at cut-off, and XOC will be the crank-angle at cut-off. Assume a lead angle of about 7° and draw OA, which, if this assumption be true, will represent the crank-angle at admission. Bisect the angle COA by the line OE, and on OE draw the valve circle. Draw the lap circle VNK. With this assumed eccentricity we find a maximum port opening of NE = .75 inch, which is larger than the conditions of the problem demand. We may then form a proportion, namely:

The actual port opening desired : the port opening with the assumed eccentricity : : probable eccentricity : assumed eccentricity.

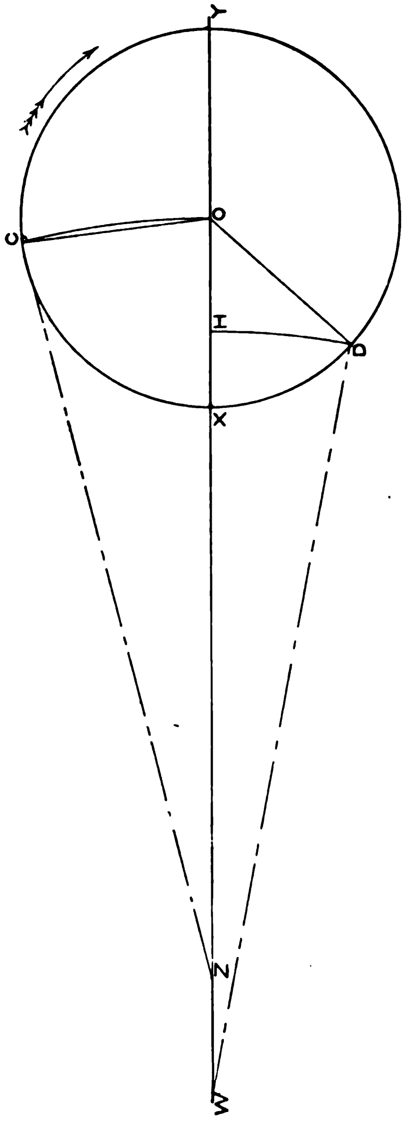

Fig. 29

Substituting the figures we have .5 : .75 : : x : 2⅛ ∴ x = the probable eccentricity; equals 1.42 inches.

Now draw on OE, a new valve circle (dotted) with a diameter equal to the required eccentricity of 1.42 inches. See Fig. 29*a*. It will cut the crank line

OC at K′, and OK′ will be the new outside lap and I′J′ will be the new lead (assuming the lead angle to be 7°). This lead I′J′ is $\frac{1}{8}$ inch, while the required lead is only $\frac{1}{16}$ inch. Now decrease the angular advance enough to correct one half of this difference, by drawing a new lap circle J″K″ of $\frac{1}{16}$ inch greater radius. This will make the valve circle cut OC at K″, so that OK″ will now be

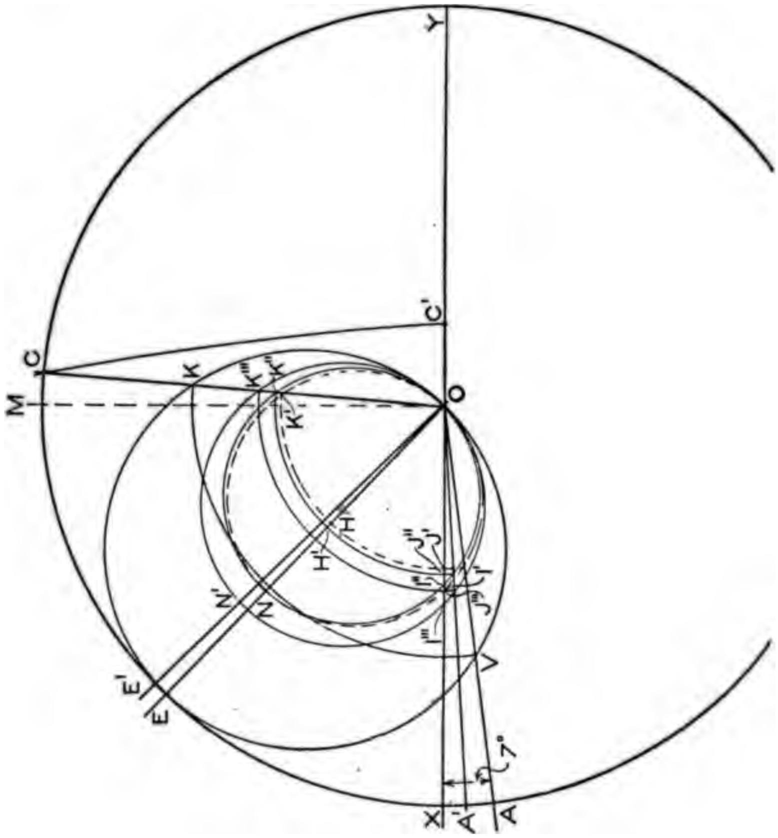

Fig. 29.

the final lap, and I″J″ the final lead, which is equal to the required $\frac{1}{16}$ inch. The lead angle is now XOA′ instead of XOA. The port opening at NH′ is $\frac{1}{2}$ inch (nearly) as required, but the change in angular advance necessitates an increase of lap if cut-off is to remain the same. This reduces the port opening by the amount HH′, so that the maximum opening is only .46 inch. By increasing the eccentricity this port opening may be increased.

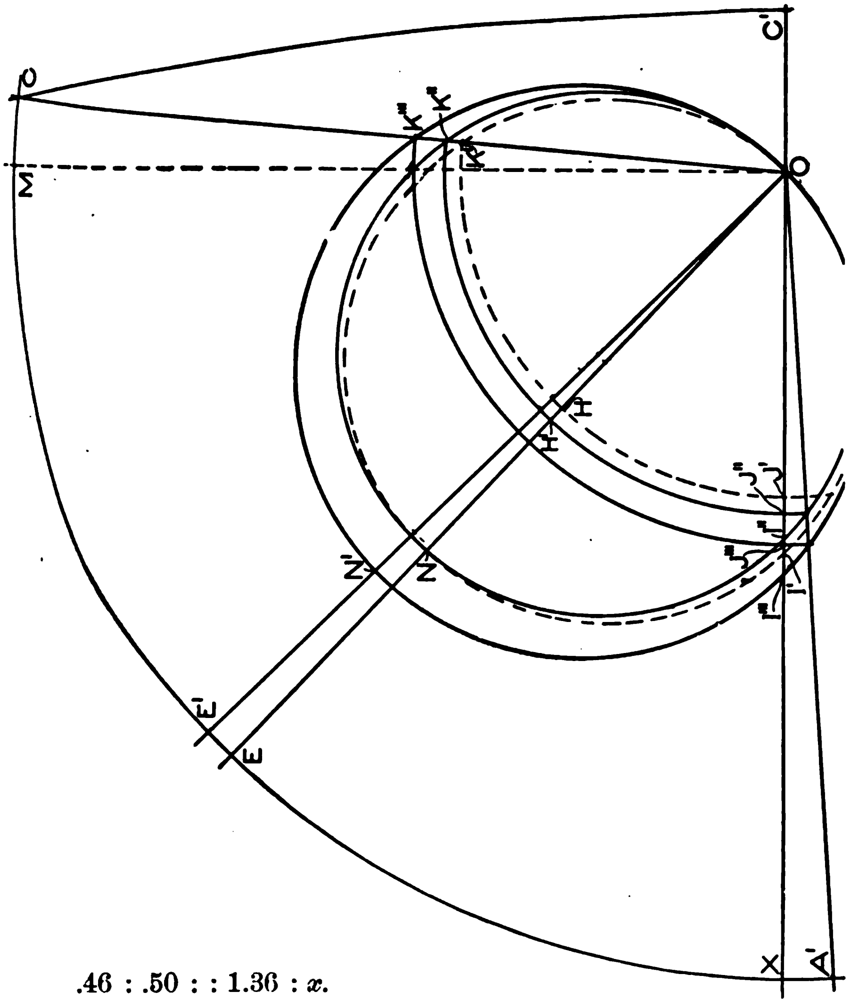

Fig. 29a. Essentials of Fig. 29 to larger scale.

.46 : .50 : : 1.36 : x.

$x = 1.48$, the true eccentricity.

Now draw the valve circle on OE' with a diameter of 1.48 inches. It will cut OC in K''' and OX in J'''. The lap will be OK''' = .97 inch, the lead will be I''' J''' = $\frac{1}{16}$ inch, the angular advance will be MOE' and the eccentricity ON'.

To sum up we have

Given:

The cut-off = **.6 stroke.**
The lead = **$\frac{1}{16}$ inch**
Max. port opening = **½ inch.**

Obtained all of the above conditions together with:

Lap = .97 inch.
Lead angle = XOA′°
Angular advance = MOE′°

Compression, release and inside laps are found as in the previous problems.

There are of course all sorts of combinations that would make up different problems, but they can all be solved in the same general way, as they are modifications of the above.

DESIGN OF THE SLIDE VALVE.

In designing a slide valve some of the variables are assumed and the others are found by means of diagrams as we have already seen. These diagrams show only the dimensions of the inside and outside laps and travel of valve; the other dimensions of the valve and seat must be calculated.

Area of Steam Pipe. Pipes that supply the steam chest should be large enough to prevent an excessive loss of pressure due to friction. If the pipes are long they should be of such size that the mean velocity of steam in them does not exceed 100 feet per second or 6,000 feet per minute. For this calculation it is usual to assume steam admitted to the cylinder during the whole stroke.

For example. Suppose an engine is 10″ × 18″, and makes 180 revolutions per minute. What is the diameter of the steam pipe?

The piston displacement or volume of the cylinder is:

$$\frac{\pi d^2}{4} \times l = \frac{3.1416 \times 10^2}{4} \times 18 = 1413.72 \text{ cubic inches.}$$

$$\frac{1413.72}{1728} = .818 \text{ cubic feet.}$$

If the engine makes 180 revolutions it would use $2 \times 180 \times .818 = 294.48$ cubic feet per minute.

The area would be $= \frac{294.48}{6000} = .04908$ sq. ft. $= 7.0675$ sq. in.

The diameter corresponding to 7.0675 square inches is 3 inches.

A three-inch pipe would be large enough, especially if the engine cut off early in the stroke.

For a very large engine cutting off early, the allowable velocity may be taken as 8,000 feet per minute instead of 6,000 feet.

Width of Steam Port. The port opening at admission should give nearly as great an area as the steam pipe in order to prevent loss of pressure due to wire-drawing, but the actual width of the port should be great enough for the free exhaust of steam. It is well to have the steam port a little larger than the area of the steam pipe, then with a port opening of .6 to .9 of the port area for admission and full port opening at exhaust, satisfactory conditions will result.

The length of the ports is usually made about .8 the diameter of the cylinder. Then in the 10″ × 18″ engine the steam ports would be 8 inches long. If the area for admitting steam is 8.0675 square inches and the length of port is 8 inches, the width will be $= \frac{7.0675}{8} =$ .8834 inch, or about $\frac{7}{8}$ inch.

The width of port opening would be about .9 × .8834 = .79506 inch or about $\frac{13}{16}$ inch.

Width of Exhaust Port. When the slide valve is at its maximum displacement, the valve overlapping the exhaust port as shown in Fig. 7 reduces the area more or less. In designing the valve, the exhaust port should be of such a width that the maximum displacement of the valve does not reduce the area of the exhaust port to less than the area of the steam port. It is not advisable to make the exhaust port too large for this increases the size of the valve and thus causes excessive friction.

The height of the exhaust cavity should never be less than the width of the steam port, and may be made much higher to advantage.

Width of Bridge. The bridge must be of sufficient width so that outside edges of the valve cannot uncover the exhaust port. The width of the steam port plus the width of the outside lap plus the width of the bridge must be greater than the maximum displacement.

The width of the bridges should be not less than the thickness of the cylinder wall in order to make a good casting.

The Point of Cut-off. In the study of Indicators, it was shown that if the point of cut-off is early, the other events are not good. If a plain slide valve is used with an automatic cut-off, the cut-off is hastened either by changing the eccentricity or by changing the angular advance. Either of these methods will accomplish the result at the expense of the compression which consequently will be earlier and excessive. Except for locomotives and high-speed engines, where compression is an advantage, the plain slide valve is not arranged to cut-off earlier than $\frac{1}{2}$ or $\frac{2}{3}$ stroke. If an earlier cut-off is desired, large outside laps are necessary. The cut-offs may be equalized by giving the head end a greater lap than the crank end, but this will cause an inequality of lead.

Lead. The lead of stationary engines varies from zero to $\frac{3}{8}$ inch according to the style of engine. An engine having high compression that compresses the steam nearly to boiler pressure, will give good results with little or no lead. If the ports are small, and the clearance large, there should be considerable lead in order to insure full initial pressure on the piston at the beginning of the stroke. Valves that open slowly require more lead than quick-acting valves.

Let us design and lay out the valve and valve seat for the following engine:

Diameter of cylinder = 10 inches.
Stroke = 18 inches.
Revolutions = 180 per minute.
Lead angle = 3°.
Cut-off to be equal at both ends and to take place at .75 stroke.
Max. port opening = .9 area of steam pipe.
Compression to be .85 of the stroke at both ends.
Length of connecting rod = 3 feet.

The piston displacement, or cylinder volume, will be $\frac{3.1416 \times 10^2}{4} \times 18 = 1413.7$ cubic inches or .818 cubic feet. If the engine makes 180 revolutions, it will use $2 \times 180 \times .818 = 294.48$ cubic feet of steam per minute. Steam pipe area $= \frac{294.48}{6000} = .0491$ square feet = 7.07 square inches.

This 7.07 square inches would also be the least possible area of the steam ports. If the length of port is made .8 the diameter of cylinder, the width will be $\frac{7.07}{8} = .88$ inches or about $\frac{7}{8}$ inch. The width of maximum port opening will be $.9 \times .88 = .792$ or nearly $\frac{13}{16}$ inch.

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ASTOR, LENOX
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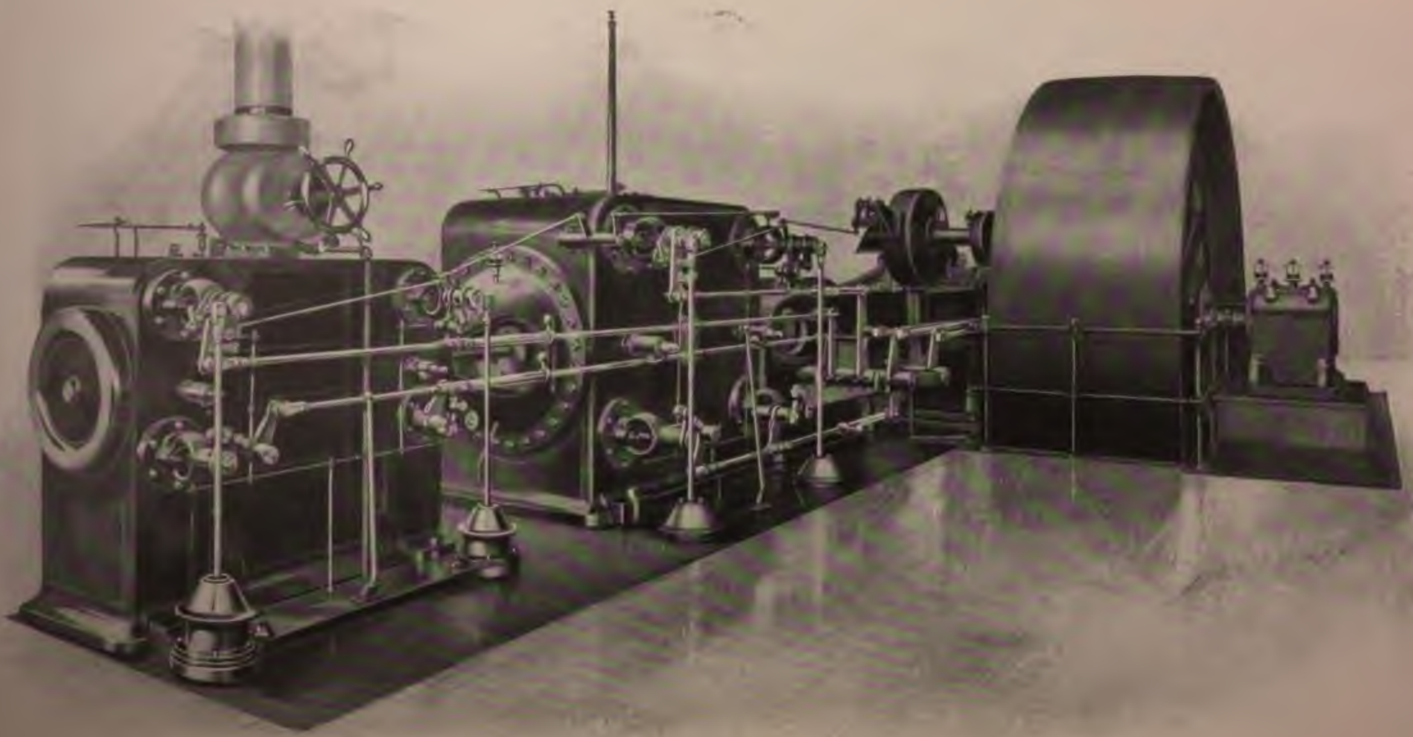


HORIZONTAL TANDEM RICE AND SARGENT ENGINE.
Providence Engineering Works.

It will be necessary to draw a separate valve circle for each end of the cylinder. First consider the head end.

The valve travel not being known, we shall lay off XY on an assumption of 6 inches travel and draw the eccentric circle as shown in Fig. 30. Lay off the lead angle XOA = 3°. Lay off XC′ = .75 of the assumed valve travel = 4½ inches. Draw the arc CC′ as previously explained and draw OC which will be the crank angle at cut-off. The radius of the arc C′C will be equal to 4 times the radius of the eccentric circle, or 12 inches, because the connecting rod is 4 times the length of the crank. Bisect the angle AOC by the line OE, and on OE draw the valve circle. OV = OK is then the outside lap, with these assumed condi-

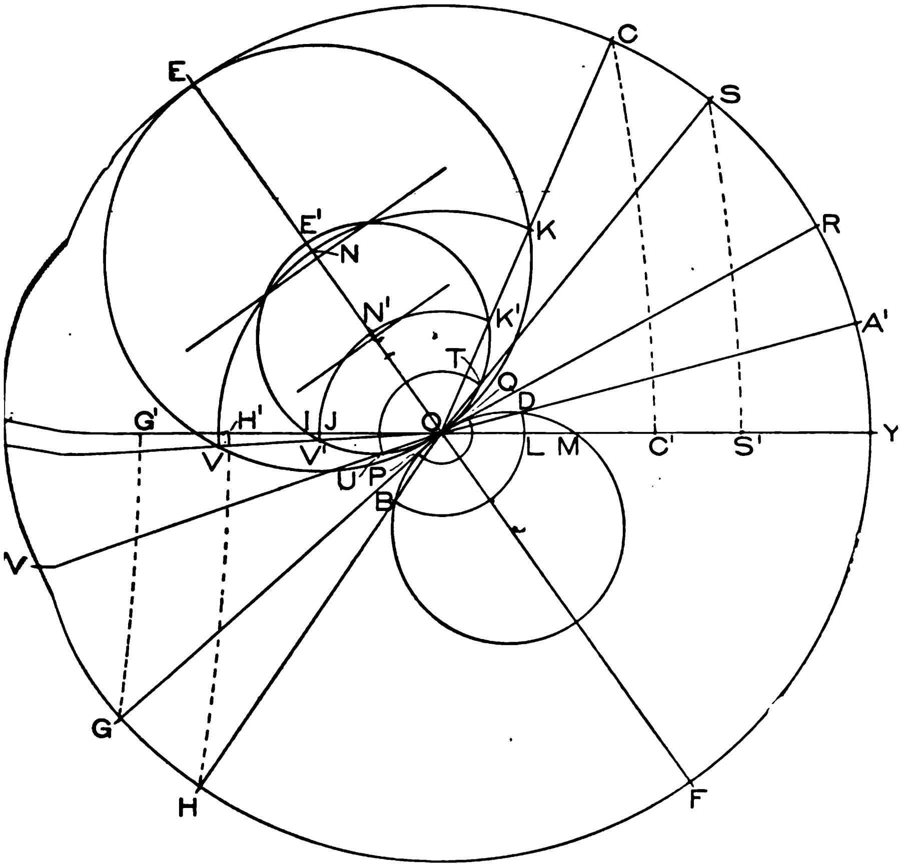


Fig. 30.

tions. Draw the lap circle; then EN will be the maximum port opening. . EN = $1\frac{7}{16}$ inches, while $\frac{13}{16}$ inch is all that is necessary. The assumed eccentricity is 3 inches, therefore the probable eccentricity $= x : 3 : : \frac{13}{16} : 1\frac{7}{16}$. $x = 1\frac{11}{16}$ inches.

Now draw a new eccentric circle with a radius of $1\frac{11}{16}$ inches and a new valve circle with OE′ = $1\frac{11}{16}$ inches as a diameter. OK′ is now the outside lap and the maximum port opening is equal to E′N′, which from actual measurement is found to be $\frac{13}{16}$ inch. The outside lap = OK′ = OV′ = $\frac{27}{32}$ inch and the lead is IJ = $\frac{3}{32}$ inch.

Produce EO to F and draw another valve circle. We shall use this valve circle to determine the outside laps and lead for the crank end of the cylinder. Since the cut-off is to be .75 of the stroke, we may lay off OH′ = OC′, and with a radius of 12 inches draw the arc HH′. Then, as already explained, OH will be the crank angle at cut-off on the return stroke. OB will be the outside lap = $\frac{19}{32}$ inch. Draw the lap circle intersecting the valve circle at D. Then ODA′ is the crank angle at admission on the return stroke and LM = $\frac{3}{8}$ inch is the lead on the crank end of the cylinder. The maximum port opening will always be greater at the crank end than at the head end because the crank end lap is less in order to get the equal cut-off. If the laps were equal, of course the port openings would be equal.

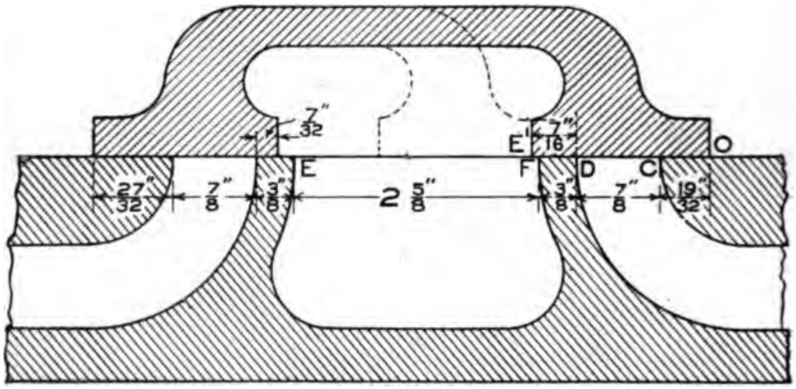


Fig. 31.

Now lay off YG′ = .85 of XY and find the crank position OG. This is the compression on the head end of the cylinder and gives an inside lap on this end of $\frac{7}{32}$ inch, which is equal to OP.

Draw the lap circle PQ, which allows us to draw through Q the crank line OR, which is the release on the forward stroke.

Lay off $XS' = YG' = .85$ of XY, and construct the crank line OS, which is the crank position at the crank end compression. OS intersects the valve circle at T, which gives $OT = \frac{7}{16}$ inch = inside lap on the crank end. Draw this lap circle, which will intersect the valve circle at U. This enables us to draw OUW, the crank angle at release, on the return stroke.

From the data determined by means of these diagrams the valve may now be laid out. For convenience let us tabulate the results obtained as follows:

Data.	Head End.	Crank End.
Cut-off, per cent of Stroke	75	75
Outside Lap	$\frac{27}{32}''$	$\frac{19}{32}''$
Inside Lap	$\frac{7}{32}''$	$\frac{7}{16}''$
Lead	$\frac{3}{32}$	$\frac{3}{8}''$
Port Opening	$\frac{13}{16}''$	$1\frac{1}{16}''$
Width of Port	$\frac{7}{8}''$	$\frac{7}{8}''$

Fig. 31 shows this valve in section. Let us begin at the end having the largest inside lap, or in this case at the crank end. Lay out the steam port $\frac{7}{8}$ inch wide, and the crank-end outside lap $= \frac{19}{32}$ inch. The bridge will be, say, $\frac{3}{8}$ inch wide. From the inner edge of the steam port, lay off the crank-end inside lap $= \frac{7}{16}$ inch. When the valve moves to the left, the point E′ will travel $1\frac{1}{16}$ inches, a distance equal to the eccentricity, and in this position of extreme displacement the exhaust port EF must be open an amount at least equal to the steam port, $\frac{7}{8}$ inch. Therefore we lay off EF equal to $1\frac{1}{16}'' + \frac{7}{8}'' = 2\frac{9}{16}''$. The inside lap overlaps the bridge nearly $\frac{1}{8}$ inch, so that we shall have to make the exhaust port opening equal to $2\frac{5}{8}$ inches. Lay off $\frac{3}{8}$ inch again for the bridge and measure back $\frac{7}{32}$ inch, equal to the head-end inside lap. The port is $\frac{7}{8}$ inch wide, and the head-end inside lap of $\frac{27}{32}$ inch completes the outline of the valve seat.

VALVE SETTING.

The principles of valve diagrams are useful in setting valves as well as in designing them. The valve is usually set as accurately as possible, and then, after indicator cards have been taken, the final adjustment can be made to correct slight irregularities.

The slide valve is so designed that the laps cannot be altered without considerable labor, and the radius of the eccentric, which determines the travel of the valve, is usually fixed. The adjustable parts are commonly the length of the valve spindle and the angular advance of the eccentric.

By lengthening or shortening the valve spindle, the valve is made to travel an equal distance each side of the mid-position. Moving the eccentric on the shaft makes the action of the valve earlier or later as the angular advance is increased or decreased.

To Put the Engine on the Center. It is usual to put the engine on center before setting the valve. First put the engine in a position where the piston has nearly completed the outward stroke, and make a mark M on the guide opposite the corner of the crosshead or at some convenient place. Also make a mark, with a center punch, on the frame of the engine near the crank disc or on the floor. With this punch mark P as a center, describe an arc C on the wheel rim, with a tram. A tram is a steel rod with its ends bent at right angles and sharpened.

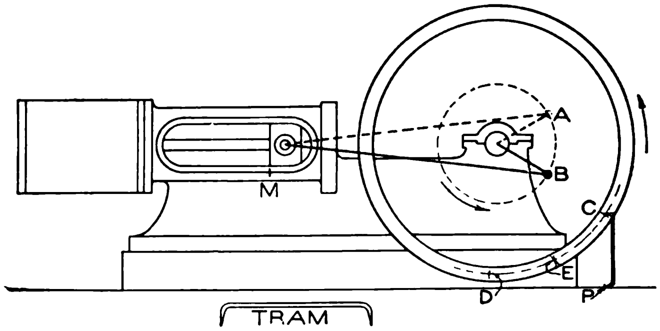


Fig. 32.

Turn the engine past the center until the mark on the guide again corresponds with the corner of the crosshead, and make another mark D on the wheel with the tram, keeping the same center. With the center of the pulley or crank disc as a center, describe an arc CD on the rim, which intersects the two arcs drawn with the tram. Bisect the arc CD on the rim, included between the two short arcs, and turn the engine until the new point E is at a distance from the point on the frame equal to the length of the tram, in which position the engine will be on the center.

The engine should always be moved in the direction in which it is to run so that the lost motion of the wrist pin and crank pin will be taken up the right way. In case the engine has been moved too far at any time, it should be turned back beyond the desired point and brought up to that point while the engine is moving the right way.

To Set the Valve with Equal Lead. Set the engine on the dead point and give the eccentric the proper angular advance. Adjust the length of the valve spindle to give the proper lead for that end. Now place the engine on the other dead point and measure the lead at that end. If the leads are unequal, correct half the error by changing the length of the valve spindle and the other half by altering the angular advance. In case the valve gear has a rocker, the length of the spindle should be such that the rocker will move as designed. The angular advance should not be changed, but the equal lead should be obtained by means of the valve spindle or the eccentric rod.

Second Method. In case it is difficult to turn an engine the following method may be used. First loosen the eccentric on the shaft and turn it around until it gives maximum port opening first at one end and then at the other. If the maximum port openings are not equal, make them so by changing the length of the valve spindle by half the difference. When the above adjustment has been made, set the engine on dead center and give the valve the proper lead by turning the eccentric on the shaft. The angular advance is thus adjusted.

To Set the Valve for Equal Cut-off. Place the engine on the dead point, give the eccentric the proper angular advance and the valve the proper lead. Move the engine forward until cut-off occurs, then measure the displacement of the crosshead from the beginning of the stroke. Continue moving the engine forward, until cut-off takes place on the return stroke and measure the displacement of the crosshead from the beginning of this stroke to this point.

In case the cut-off is earlier at the crank than at the head-end, the valve spindle is too short. Adjust the length of the spindle so that the inequality will be corrected. Now set the engine on the dead point again and give the valve the proper lead by means of the eccentric. By repeating the process, making slight changes, the desired result will be obtained.

MODIFICATIONS OF THE SLIDE VALVE.

The ordinary slide valve is suitable for small engines; but for large sizes some method must be employed to balance the steam pressure on the back of the valve. With large valves, such for instance as those of locomotives or large marine engines, a great force is exerted by the steam, and the valve is forced against its seat so hard that a large amount of power is necessary to move it. This excessive pressure causes the valve to wear badly and is a dead loss to the engine. The larger the valve, the greater this loss will be.

Piston Valve. To prevent excessive pressure on the back of the valve, the piston valve is commonly used, especially in marine engines. This valve consists of two pistons, which cover and uncover the ports in precisely the same manner as the laps of the plain slide valve. These pistons are secured to the valve stem in an approved manner and are fitted with packing rings.

The valve seat consists of two short cylinders or tubes accurately bored to fit the pistons of the valve. The port openings are not continuous as in the plain slide valve, but consist of many small openings, the bars of metal between these openings preventing the packing rings from springing out into the ports.

Steam may be admitted to the middle of the steam chest and exhausted from the ends or vice versa. With the former method, the live steam is well separated from the exhaust, and the valve-rod stuffing box is exposed to exhaust steam only. This is a good arrangement for the high-pressure cylinder; if used for a cylinder in which there is a vacuum, air may leak into the exhaust space through the valve-rod stuffing box. With this arrangement the steam laps must be inside and the exhaust laps on the outside ends.

The piston valve may be laid out and designed by means of the Zeuner diagram just as if it were a plain slide valve, and the action is the same except that it is balanced so far as the steam pressure is concerned; the power to drive it being only that necessary to overcome the friction due to the spring rings.

Fig. 33 shows a section of the piston valve and the high-pressure cylinder for one of the engines of the U. S. S. "Massachusetts." This valve consists of two pistons connected by a sleeve through which the valve rod passes. This valve rod is prolonged to a small balancing piston, placed directly over the main valve. The upper end of the balancing cylinder does not admit steam, so that the steam pressure below the balancing piston will practically carry the weight of the piston valve, thus relieving the valve gear and making the balance more nearly complete.

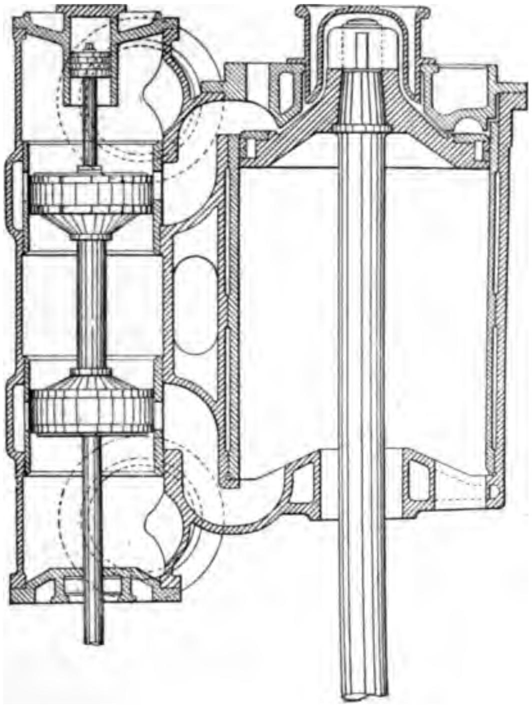


Fig. 33.

Double-Ported Valve. Sometimes it is impossible to get sufficient port opening for engines of large diameter and short stroke, especially those having a plain slide valve with short travel.

This difficulty may be overcome by means of the double-ported valve shown in Fig. 34. It is equivalent to two plain slide valves, each having its laps. The inner valve is similar to a plain slide valve except that there is communication between the exhaust space and the exhaust space of the outer valve. Each passage to the cylinder has two ports; a bridge separates the exhaust of the outer valve from the steam space of the inner valve, and the outer valve is made long enough to admit steam to the inner valve.

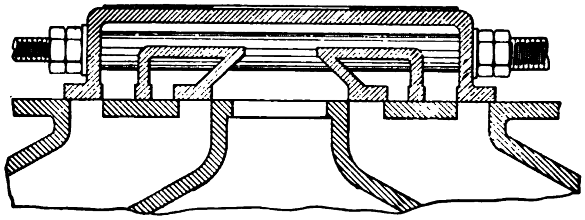


Fig. 34.

This valve may be considered as equivalent to two equal slide valves of the same travel, each having one-half the total port opening. To admit the same amount of steam as a plain slide valve, the double-ported valve requires but half the valve travel; this is advantageous in high-speed engines.

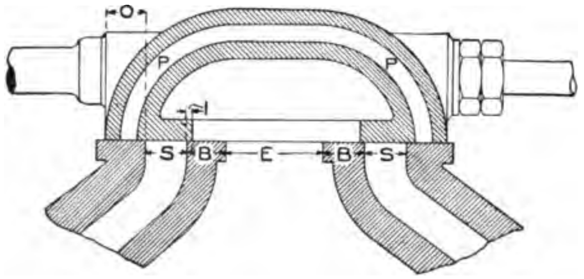


Fig. 35.

To balance the excessive steam pressure, the back of the valve is sometimes provided with a projecting ring which is fitted to a similar ring within the top of the valve chest. These rings are planed true, and fit so that steam is prevented from acting on the back of the valve. The space inside the rings is sometimes placed in communication with the condenser.

The Trick Valve. The defect of the plain slide valve, due to the slowness in opening and closing, is largely remedied in the trick valve, which is so made that a double volume of steam enters during admission. Thus a quick and full opening of the port is obtained with a small valve travel.

In Fig. 35 the valve is shown in mid-position. It is similar to a plain slide valve except that there is a passage PP through it. It has an outside lap O and an inside lap I. The seat is raised and has steam ports SS, bridges BB, and exhaust port E. If the valve moves to the right a distance equal to the outside lap plus the lead, it will be in the position shown in Fig. 36. Steam will be admitted at the extreme left edge of the valve just the same as though it were a plain slide valve; also, since steam surrounds the valve it will be admitted through the passage as shown in Fig. 36.

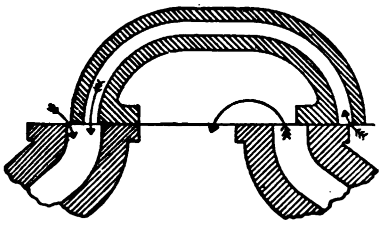


Fig. 36.

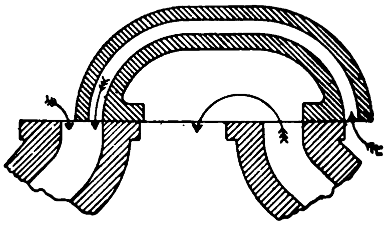


Fig. 37.

If the lead is the same as for a plain slide valve, $\frac{1}{16}$ inch for instance, this valve would give double the port opening, that is $\frac{1}{8}$ inch, when the valve was open a distance equal to the lead.

Fig. 37 shows the valve when it is in its extreme position to the right and the port is full open to steam.

Piston valves are also made with a passage similar to that of the trick valve for double admission. The valve used with the Armington and Sims engine is perhaps the best example.

Balanced Valves. Since there is a wide difference between the pressure of admission and exhaust, there must always be a great pressure acting upon the valve, causing it to run hard and wear excessively. The greater the steam pressure, the lower the pressure at exhaust and the larger the valve, the greater this pressure will be.

Piston valves are commonly used on the high and intermediate cylinders of triple-expansion engines, and if well made and fitted with spring rings, should not leak. Small piston valves are often made without packing rings; but even if they fit accurately when new, they soon become worn and cause trouble.

The double-ported valve, the trick valve, and others often have some device for relieving the pressure, such as a bronze ring or cylinder, fastened to the back of the valve. This ring is pressed by springs against a finished surface of the valve chest cover, and the space thus enclosed by the ring may be connected to the exhaust. There are numerous devices for balancing valves, but they are usually more or less expensive and are liable to cause trouble from leakage.

STEPHENSON LINK MOTION.

One of the earliest, and at present one of the most common mechanisms for reversing engines, or changing the ratio of expansion, is the Stephenson link motion, shown in Fig. 38. This illustration is taken from the drawings of a recent battleship engine, and may be considered the typical arrangement of the Stephenson gear as applied to marine practice.

The two eccentrics E and E', whose centers are at C and C, respectively, are shown in their relative positions when the crank OA is at dead center. The eccentric rods R and R' are connected by forked ends to the link pins H and G. The link consists of two curved bars bolted togethe rin such a manner that they may slide by the link block N. On the link are three sets of trunions; the two outer ones, or link pins, are fitted into the forked end of the eccentric rods, and the middle one, known as the saddle pin, is fitted into the end of the drag links FM.

The valve stem has, at its lower end, a pivoted block N, called the link block, provided with slotted sides through which the links can slide from right to left. The reverse shaft, or rock shaft, K, here shown in full gear "forward," may be turned until F moves over to B; in this position the link will be pushed across the link block, and the valve will get its motion from the rod R' instead of from R as before. The link in this position would be full gear "astern."

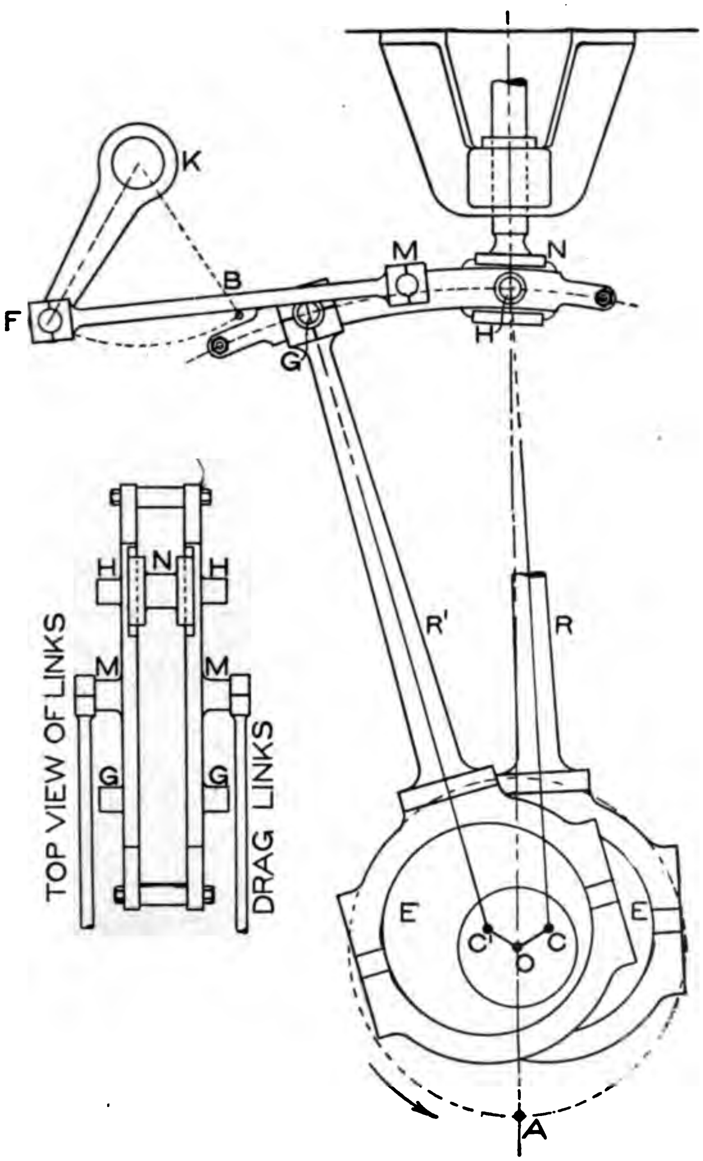


Fig. 38.

In all large engines, such as marine, the reverse shaft is turned by power, but in smaller engines, such as locomotives, the engineer can turn the shaft by means of a lever.

When set full gear forward, as in Fig. 38, the valve admits steam to the crank end of the cylinder, and the crank revolves as shown by the arrow. As the crank turns, both eccentrics impart motion to the link, but the "go ahead" link pin H approximately coincides with the link block, so that nearly all its up-and-down motion is transmitted to the valve stem, while the "go astern" eccentric exerts but little effect upon the link block. Moving the drag links over to the extreme right reverses all these conditions by bringing the other link pin under the link block. In this position, steam will be admitted to the other end of the cylinder, and the engine will run in the opposite direction. This will be clearly seen by referring to Fig. 38.

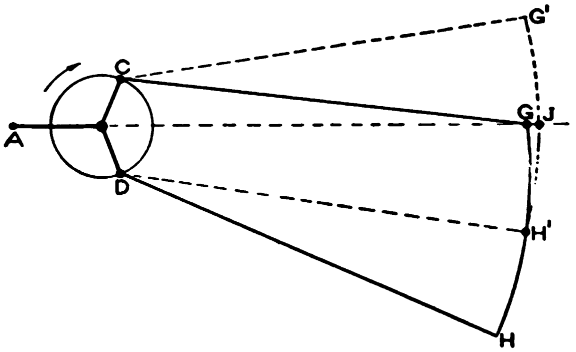


Fig. 39.

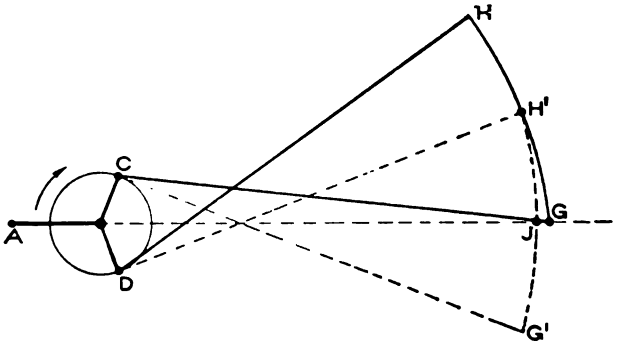


Fig. 40.

When at full gear, either forward or backing, the valve moves as if there were really but one eccentric, while at intermediate points its motion is the result of the combined influence of both eccentrics, one tending in a measure to counteract the other. The effect of this is to shorten the valve travel the same as if the valve were driven by a new eccentric having less throw than either of the other two.

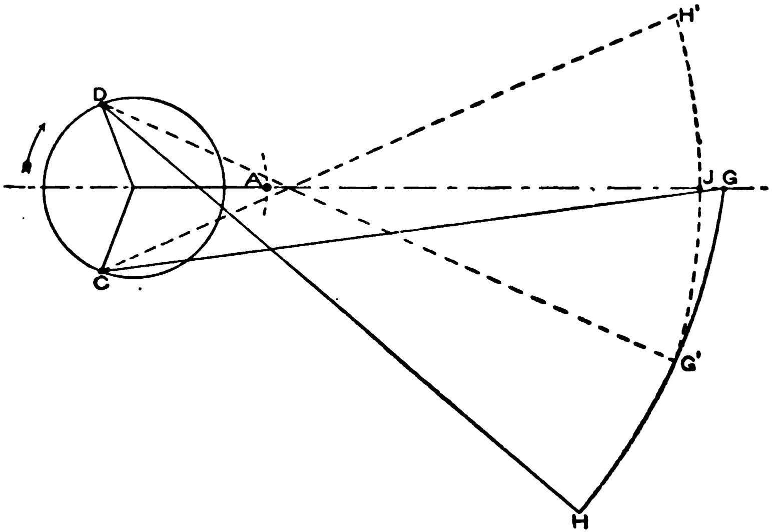


Fig. 41.

Decreasing the valve travel causes cut-off to occur earlier compression is earlier, release later, and the lead is reduced somewhat. If every point of the link moved in the arc of a circle when the drag link is shifted, the lead would not alter; but, since the eccentric rods about which each end swings are centered at different points, C and C′, this is impossible.

Figs. 39 and 40 show the two principal ways of arranging the eccentric rods of a Stephenson gear. The first is said to have "open rods", the second "crossed rods"; referring to whether the rods are crossed or open when both the eccentrics face the link. It can easily be seen that when the eccentrics shown in Fig. 39 have turned through 180° they will be in the position shown in Fig. 41, but this is the same arrangement as before and is "open" rods. The full lines show the positions in full gear forward, while the dotted lines indicate the positions in mid gear. With open rods it will be seen that when at full gear the link block is at G, and that if, without turning the crank, the link is shifted to mid gear, then the link block moves to J, Fig. 39, and the valve must consequently be moved toward the right an amount equal to GJ, thereby increasing the lead on the crank end of the cylinder. With crossed rods, moving the link from full to mid gear moves the link block from G to J, Fig. 40, thus reducing the lead. It follows then that open rods give increasing lead from full toward mid gear, and that crossed rods give decreasing lead. With crossed rods there will be no lead when in mid gear. It will be apparent that the shorter the rods the greater this increase or decrease will be.

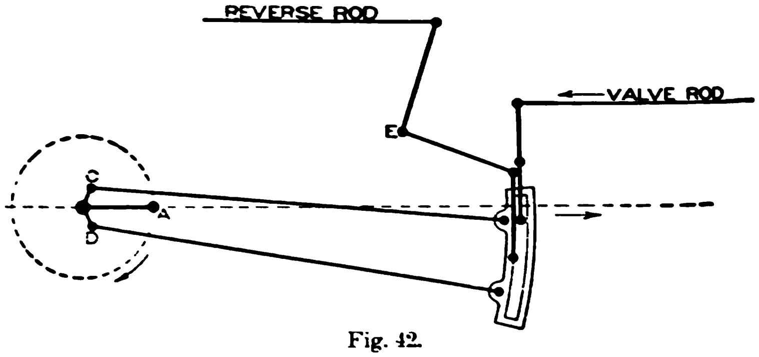


Fig. 42.

Nearly all marine engines, and some English locomotives, have their link blocks carried directly on the valve rod. American locomotives commonly use a rocker, one end of which carries the link block while the other moves the valve rod. This arrangement indicated in Fig. 42 makes it possible to place the valve and steam chest above the cylinder. The position of the crank for the same valve position is just opposite that shown in Fig. 39 because the rocker reverses the valve motion; this gives an arrangement of crank and eccentrics that is identical with that indicated in Fig. 41 and the rods, although apparently crossed, are in reality of the open rod arrangement, giving increasing lead toward mid gear. A rod from the bell-crank lever on the reverse shaft E, leads back to the engineer's cab and connects with the reverse lever. This ever moves over a notched arc, and may be held by a latch in any one of the notches, thus setting the link in any position from mid gear to full gear, either forward or back.

The Stephenson link is designed to give equal lead at both ends of the cylinder; but to accomplish this, the radius of the link arc (that is an imaginary line in the center of the slot) must be equal to the distance from the center of this slot to the center of the eccentric. In Fig. 38 the radius of the link arc is equal to CH and C'G.

Exact quality of lead is not essential, and the radius of the link arc is sometimes made greater or less than stated above in order to aid in equalizing the cut-off; but the change should never be great enough to affect the leads.

Stephenson originally intended to use the link simply as a reversing gear, but soon found, however, that at intermediate points between the two positions of full gear, it would serve very well as a means of varying the expansion and cut-off. Very soon the link came to be used not only on locomotives and marine engines, but on stationary engines as well, in connection with the reverse shaft which was under the control of the governor. The mechanism proved to be too heavy to be easily moved by a governor and it has gradually fallen into disuse on stationary engines excepting as a means of reversing.

In marine practice, the variable expansion feature is of little value, for marine engines run under a steady load and the link is set either at full gear or at some fixed cut-off. For locomotives, however, the variable expansion is nearly as important as reversing. Locomotives are generally started at full gear, admitting steam for nearly the entire stroke, and then exhausting it at relatively high pressure. This wasteful use of steam is necessary to furnish the power needed in starting a train. After the train is under way, less power is required per stroke, and the link is gradually moved toward mid gear, or "notched up" by the engineer, thus hastening the cut-off; the expansion is increased and the power is reduced in proportion to the load.

As the cut-off is changed, it is desirable to maintain an approximately equal cut-off at each end of the cylinder; this can be secured in the Stephenson gear by properly locating the saddle pin and the reverse shaft. When used without a rocker, as in

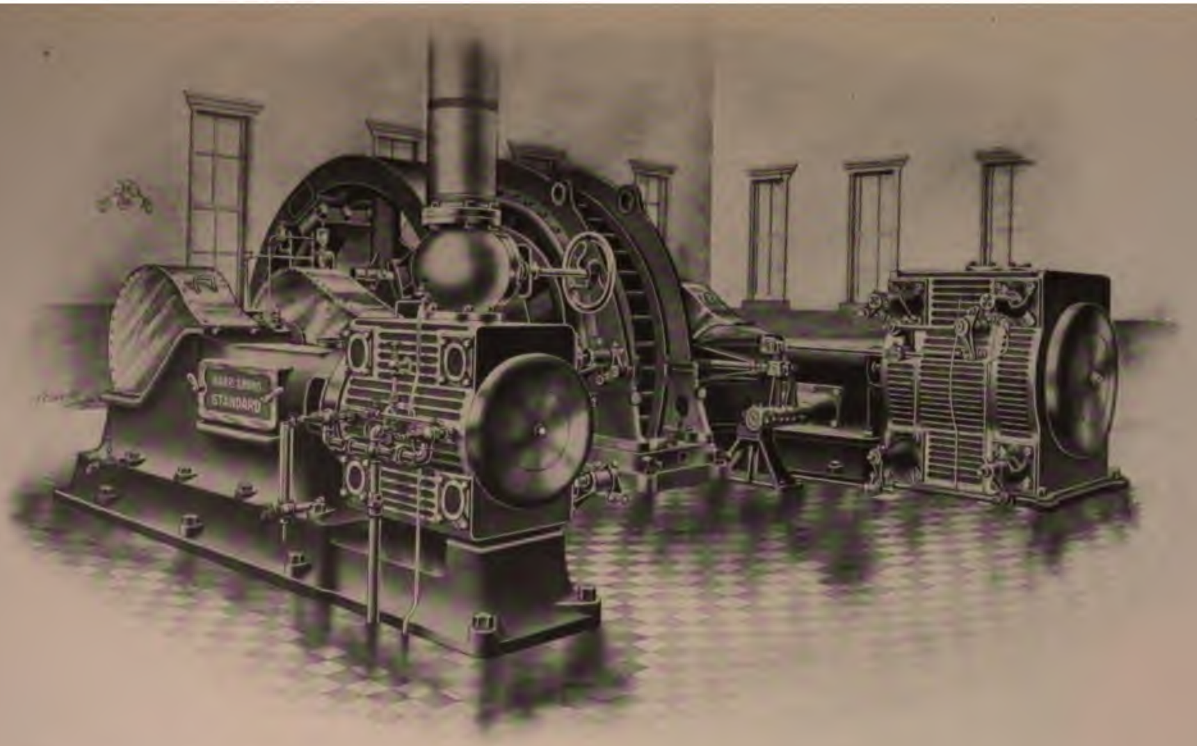
Fig. 38, the saddle pin should be on the arc of the link or slightly ahead of it. When used with a rocker, the saddle pin should be behind the link arcs, and to give symmetrical action for forward and backward running, it should be opposite the middle of the arc, that is, equally distant from each link pin.

The Stephenson link cannot be designed directly from the Zeuner diagram, but a systematic investigation can be made by using a wooden model of the proposed link. This can be mounted on a drawing board, and the effect of changing the position of pins and the proportions of rods and levers can be determined without difficulty. By a system of trials a combination can be found best suited to obtain the desired results. Moreover, a model makes it possible to measure directly the slip of the link block along the link. This slip should be kept as small as possible to prevent rapid wear. It can be controlled to some extent by properly locating the link pins, by avoiding too short a link, and by choosing a favorable position for the reverse shaft.

The Gooch Link. Another form of link motion, known as the Gooch Link, is illustrated in Fig. 43. It has been extensively used on European locomotives, although it is gradually being replaced by a type of valve gear known as the Walschaert, which will be described later.

The Gooch link has its concave side turned toward the valve instead of toward the eccentric. The radius of curvature of the link is equal to AB, the length of the radius rod. The link is stationary and the link block slides in the link. The engine is reversed by means of the bell-crank lever on the reverse shaft E which shifts the link block instead of the link, as is the case with the Stephenson. The link is suspended from its saddle pin M, which is connected by a rod to the fixed center F, so that the link can move forward and back as the eccentricity is changed, or it can pivot about its saddle pin as the eccentrics revolve.

Since the radius of the link arc is equal to AB, it is apparent that the block can be moved from one end of the link to the other, that is, from full gear "forward" to full gear "back" without moving the point A, which is on the end of the valve rod. The lead then is constant for all positions of the block, and the distribution of steam for locomotives is slightly preferable to that



GENERAL CONSTRUCTION FLEMING CROSS COMPOUND FOUR-VALVE ENGINE DIRECT-CONNECTED STYLE, GENERAL VIEW.
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obtained by the Stephenson; but the gear is more complicated and requires nearly double the distance between shaft and valve stem.

The variable lead is perhaps a slight advantage to the locomotive, which is a slow-speed engine in starting, thus requiring but little lead. As the speed increases, and the link is "notched up", the lead is increased as the cut-off is shortened, and at high speed we have a large lead. With the Gooch link, the lead can be set for the average running speed, and although a little too great for good work at slow speed, it is a matter of small consequence, because the engine runs at slow speed but a very small fraction of the time it is in service, and the loss due to large lead at slow speed is of no consequence whatever in a day's run.

Several other link motions have been used; but at the present time probably more Stephenson link motions are used than all the other forms of reversing gear combined, and when a "link motion" is mentioned, the Stephenson is usually meant unless otherwise specified.

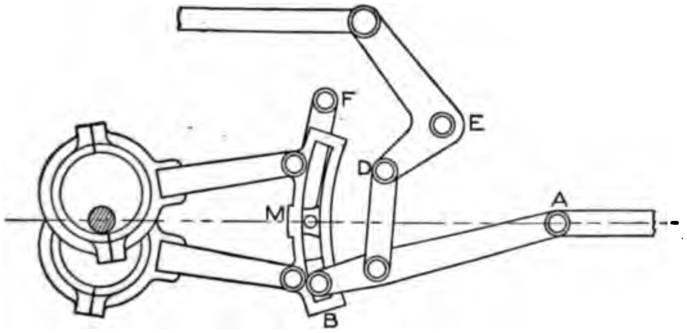

Fig. 43.

RADIAL VALVE GEARS.

In general, it would be desirable to have precisely similar steam distribution at each end of the cylinder, and it would often be of great advantage with an expansion gear like the Stephenson, if the cut-off could be shortened without changing any other event of the stroke. A Stephenson gear can be made to maintain equality of lead for both ends of the cylinder as the cut-off is shortened, but we have seen that in so doing, the lead of both ends is either increased or diminished according as the link is arranged with "open rods" or "crossed rods". Moreover, the compression is hastened by bringing the link to mid-gear, all of which in many instances is undesirable.

This disadvantage of the Stephenson link motion lead to the design of the so-called "Radial Valve Gears", many of which are so complicated as to be impracticable, but all of which obtain a fairly uniform distribution of steam.

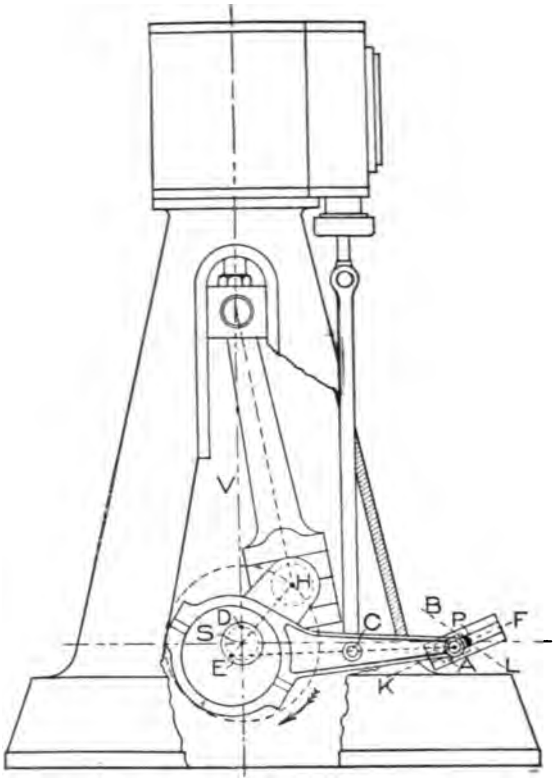

Fig. 44.

Hackworth Gear. The essential features of the Hackworth Gear are indicated in outline in Fig. 44. In this figure, S is the center of the shaft, and the eccentric E is set 180° from the crank SH. At the right-hand end of the eccentric rod EA, is pivoted a block which slides in a straight, slotted guide. The guide remains stationary while the engine is running, but can be turned on its axis P, to reverse the engine or change the cut-off. P is a pivot, located on the horizontal through S in such a position that DP = EA. If these two distances are equal, A will coincide with P when the crank is at either dead point and the slotted guide may be turned from "full gear forward", as shown in the figure, through the horizontal position to "full gear backing", as shown by the line BL, without moving the valve. Therefore the leads are constant for all positions of the guide. The valve rod running upward from C, connects with the valve stem which it moves in a straight line. The valve stem is made just long enough to equalize both leads, and if the point C has been properly chosen, the two cut-offs will be very nearly equal for all grades of the gear.

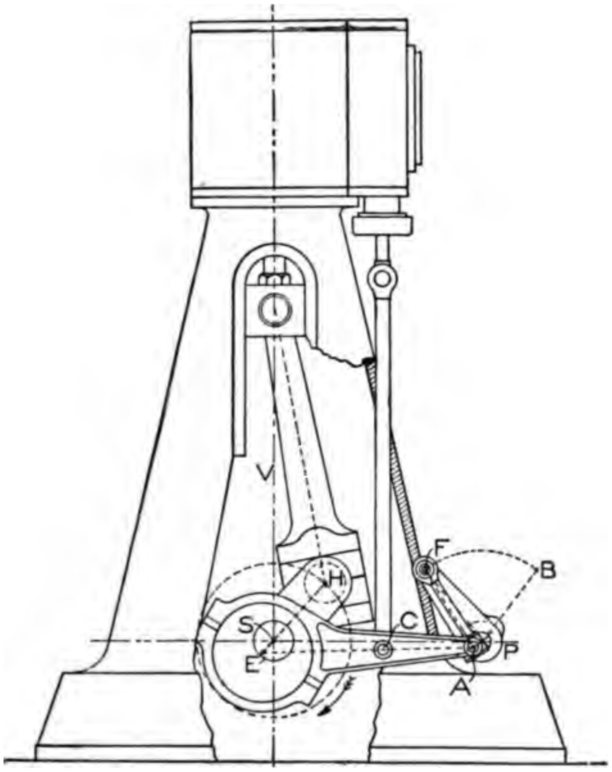

Fig. 45.

A somewhat better valve action is obtained by slightly curving the slotted guide, with its convex side downward. This gear is sometimes used on marine engines and on small stationary engines.

Marshall Gear. The most objectionable feature of the Hackworth gear, is the slotted guide, for the sliding of the block causes considerable friction and wear. The Marshall gear, shown in outline in Fig. 45, is designed to obviate this feature. The point A moves in the desired path by swinging on the rod FA about F as a center. While the engine is running, the lever FP remains stationary, but can be turned on its axis P to reverse the engine, or change the cut-off. The pivot P, is located precisely as in the Hackworth gear, and the lever FP can be turned from "full gear forward", as shown in the figure, to "full gear backing", as shown by the line BP, intermediate positions give different cut-offs as with the Hackworth gear. Since FA is made equal to FP, the point A will always swing through P, no matter where F may be, and will coincide with P, when the engine is on dead center. The leads therefore will remain constant, as in the preceding case.

The Marshall gear is sometimes made with C at the right of A on a prolongation of the line EA. In this case if the same kind of valve is to be used, the eccentric E must move with the crank instead of 180° from it. The Marshall gear is frequently used on marine engines, the one eccentric being simpler than the two required by the Stephenson.

Joy Gear. Perhaps the most widely known, and certainly one of the best radial gears is the Joy, outlined in Fig. 46. It is frequently used on marine engines and on some English locomotives. No eccentrics are used, the valve motion being taken from C, a point on the connecting rod. H is a fixed pivot supported on the cylinder casting. The lever ED has a block pivoted at A, which slides back and forth in a curved slotted guide. The guide and the lever PF are fastened to the reverse shaft P, and by means of a reverse rod leading off from F, can be turned from full gear forward, as shown, to full gear backing when the pin F moves over to B. Motion is transmitted to the valve stem by means of the radius rod EG. The proportions are such that when the crank is on either dead point, the pivot of block A coincides with P, so that the curved guide may then be set in any position without moving the valve; therefore the leads are constant. This gear gives a rapid motion to the valve when opening and closing and a more nearly constant compression than the Stephenson gear, and the cut-off can be made very nearly equal for all grades of the gear. Its many joints cause wear and its position near the crosshead, makes a careful inspection of the crosshead and piston exceedingly difficult while the engine is running.

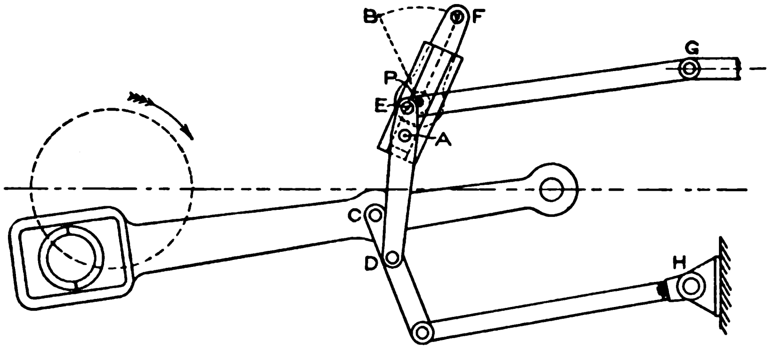

Fig. 46.

Walschaert Gear. This radial valve gear, although seldom seen in the United States, is the valve mechanism most commonly used on locomotives built on the continent of Europe. Like all other radial gears, it gives constant lead, and a distribution of steam very nearly alike for each end of the cylinder. In this respect it is superior to the Stephenson link, and gives without doubt better economy, but its mechanical construction is complicated, and not well adapted to the American type of locomotive. Fig. 47 illustrates this type of gear. S is the center of the driver axle. The crank pin K has forged on its center end an arm KE, on which the pin E is fixed. This arm lies parallel to the plane of the driving wheels, and being fixed to the crank-pin, turns with the wheel, allowing the connecting rod to pass between it and the driving wheels. In this manner the point E moves around S in a circle, and moves the rod EH back and forth just as if it were an eccentric. It is so made that ES is perpendicular to the crank KS, and therefore the action of the pin E is equivalent to an eccentric with no angular advance.

This arm reaching back from the outer end of the crank pin is one of the most objectionable features on the construction, and is sometimes replaced by the regular type of eccentric put on the shaft between the driving wheels.

The eccentric rod EH causes the box link HP to oscillate on fixed trunions P. This link has a groove curved to a radius equal to GD, the length of the radius rod. A block pivoted at G, on one end of the radius rod, is free to move up or down in this groove. The valve derives its motion from C, a pivot on the floating lever CA. Point A receives motion from the crosshead; point D from the eccentric and the curved link; and a combination of these two imparts motion to C (which can slide only along the dotted line). A bell-crank lever pivoted above the link shifts the mechanism from "full gear forward" when F is moved to B, thus raising G above the link pivot or saddle pin.

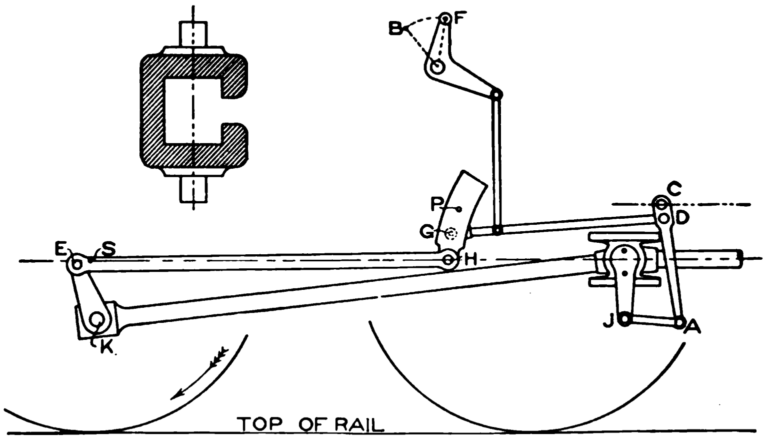

Fig. 47.

ADJUSTABLE ECCENTRICS.

The position of an eccentric for a plain slide valve is 90° plus the angular advance ahead of the crank, in the direction in which the engine is to turn. Thus A, Fig. 48, is correctly placed, relative to the crank C, if the engine turns right-handed. For running in the opposite direction, the position of the eccentric is at D. Some engines are provided with a reversing mechanism which causes the eccentric to shift from A to D, either along the arc

ABD, or along the straight line AED. Such engines provide, not only for reversing, but for changing the cut-off as well If the eccentric moves on the arc to OB, the angular advance is

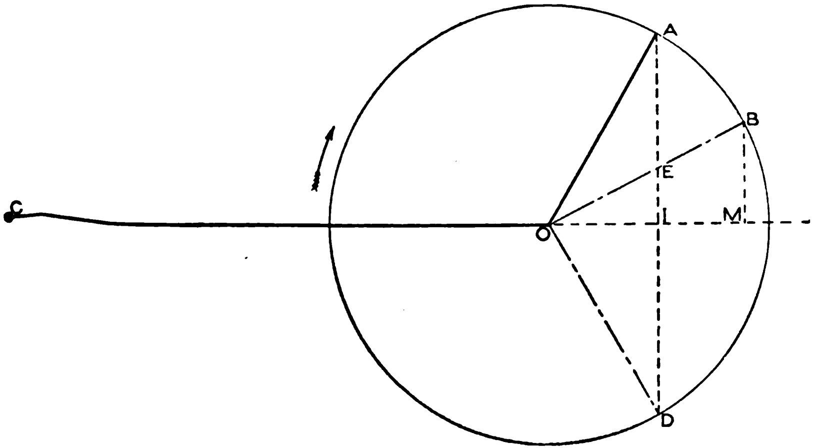

Fig. 48.

increased and all the events of the stroke are hastened as well as the cut-off, but the travel of the valve is not changed.

Zeuner's diagram, Fig. 49, is lettered to correspond with Fig. 48, and shows the effect of changing the angular advance from FOA to FOB. If OK represents the lap, the crank angle at cut-off will decrease from HOK to HOL, and the lead will increase very much, viz., from GF to HF. If the eccentric is shifted on the straight line to E (Fig. 48), a different valve motion will result. The angular advance is increased as before, so that all the events are hastened, but the eccentricity is now only OE instead of OB and the valve travel is consequently reduced. Zeuner's diagram for this case, Fig. 50, shows a decrease in crank angle at cut-off from IOM to ION, and no

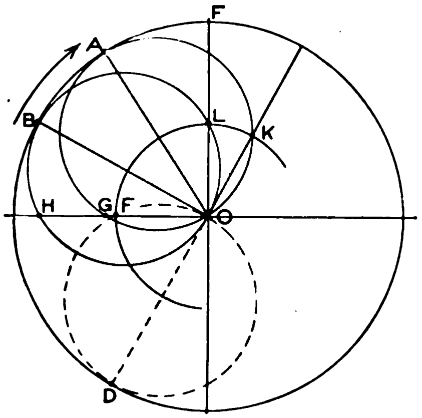

Fig. 49.

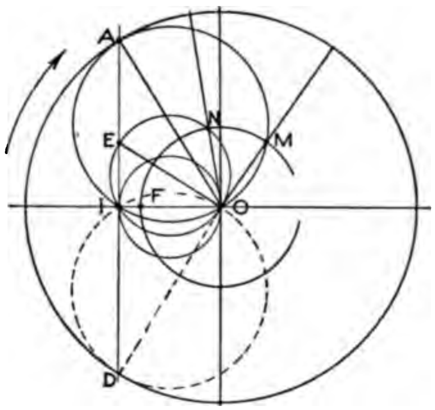

Fig. 50.

change in the lead IF. Let us consider the eccentric position OA, Fig. 48. In this position OI represents the displacement of the valve from mid-position when the engine is on center. If the eccentric moves to OB, the displacement will be OM, which is greater, showing an increase in lead equal to IM, but if the position is OE instead of OB, the displacement from mid-position will be OI as before. It is evident that the eccentric can move on the straight line from A to D, without changing the lead, while the decreased valve travel will result in an earlier cut-off. If the

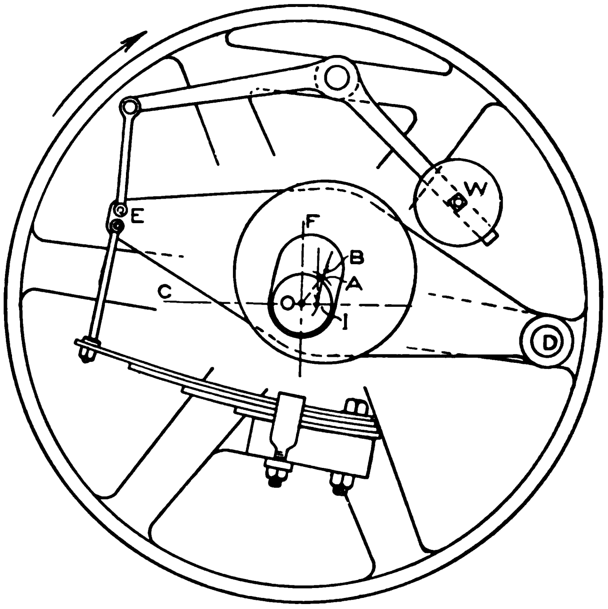

Fig. 51.

shifting eccentric is to be used for an automatic cut-off, as in the various types of fly-wheel governor, the curved path is not desirable on account of excessive lead at short cut-off, but if it is to be used only as a means of reversing, it is preferable to the straight line.

All fly-wheel governors operate by shifting the eccentric, either to change the angular advance, the travel of the valve, or both. Fig. 51 illustrates the principle of a governor arranged to give decreasing lead, but as these mechanisms are described in the Steam Engine—Part I, under the head of governors, a further discussion will not be given here.

The device shown in Fig. 52 is often used for reversing engines of small launches. The eccentric E is loose on the shaft between a fixed collar G, and a hand wheel H. A stud projecting from the eccentric passes through a curved slot in the disc of the wheel, and can be clamped by a hand nut F. When running forward with the crank at C, the center of the eccentric is at A, and the nut clamped at F. To reverse, steam is shut off, and when the engine stops, the nut F is loosened, and then moved to B and clamped; or after F is loosened, the wheel, shaft, crank and propeller are turned over by hand until B strikes the stud at F, where it is clamped. The engine will then run astern.

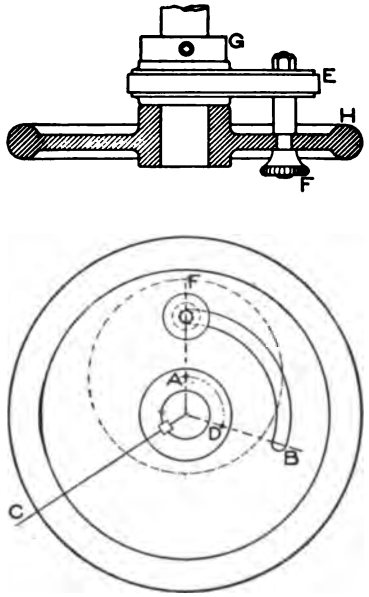

Fig. 52.

To study the application of the Zeuner diagram to this form of mechanism turn again to Fig. 51. If OA is the desired eccentricity for a normal position of the governor, the perpendicular distance of A from OF is made equal to the lap OI, plus the desired lead. Pivot D is then located equally distant from A and I. Zeuner's diagram for this gear, drawn to an enlarged scale, is shown in Fig. 53. The angular advance F'OB is laid back toward OC. OB is the maximum eccentricity; OI, the lap or the desired least eccentricity. An arc, with proper radius, described through B and I shows the path of the eccentric. If the eccentric moves in to A, the crank angle at cut-off is decreased from COD to COE, and the lead decreased from FI to GI. A slight decrease in lead is not objectionable, since the speed is not allowed to increase more than two or three per cent; and further, as the lead increases, compression decreases, so that one influence helps to counteract the other. The decrease in maximum port opening from BH to AK is unavoidable, but it is permissible, since it occurs only when the load decreases, and when less steam should be admitted to the cylinder.

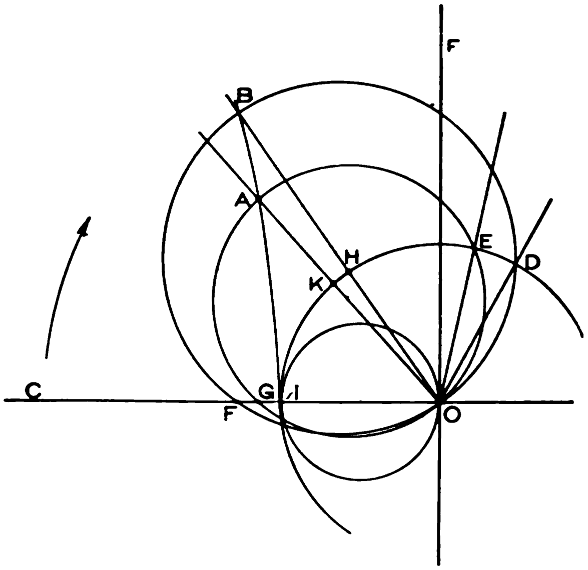

Fig. 53.

DOUBLE VALVE GEARS.

It has been shown in the preceding discussion, that a plain slide valve under the control of a gear that gives a variable cut-off, such as a shifting eccentric or a link motion, will not give a satisfactory distribution of steam at short cut-off owing to execessive compression, variable lead, or early release. These difficulties are overcome in a measure by the use of the radial gear; and also by the use of two valves instead of one. The main

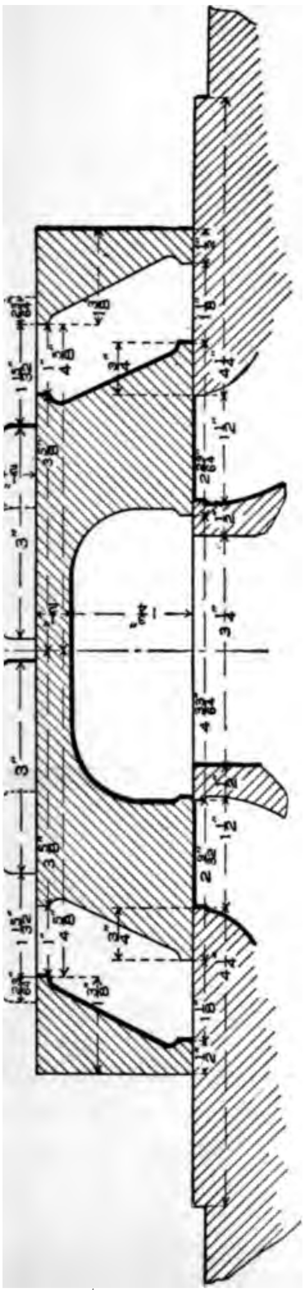

Fig. 54.

valve controls admission, release, and compression ; the other, called the cut-off valve, regulates the cut-off only, which may be changed without in any way affecting the other events of the stroke. This cut-off valve may be placed in a separate valve chest, or it may be placed on the back of the main valve.

Meyer Valve. The most common form of double valve gear is the Meyer Valve, Fig. 54. The cut-off valve is made in two parts and works on the back of the main valve. The two parts are connected to a valve spindle with a right- and left-hand thread,,so that their positions may be altered by rotating the valve spindle.

A swivel joint is usually fitted in the valve-spindle between the steam chest and the head of the valve rod, and the valve spindle prolonged into a tail rod which projects through a stuffing box on the head of the steam chest. See Fig. 55. The end of this tail rod is square in section and reciprocates through a small hand wheel by means of which it can be rotated while the engine is running, whatever the position of the valve may be.

Each valve is under the control of a separate eccentric. The eccentric moving the main valve is usually fixed, while the cut-off valve eccentric may be under the control of a governor. Since a slight compression is desired, the main valve is set to give late cut-off. This will give late release and late compression, and allow a wide range of cut-off for the cut-off valve. With this gear, lead, release, and compression are entirely independent of the ratio of expansion, and the cut-off is much sharper, because the cut-off valve, when closing the ports, is always moving in a direction opposite to that of the main valve. The valve may be designed by means of Zeuner's diagrams.

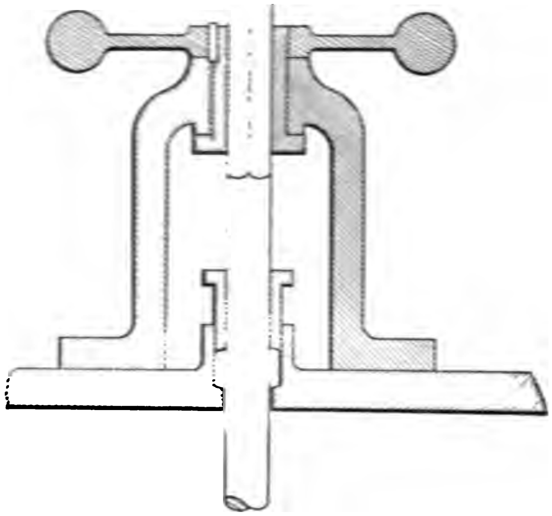

Fig. 55.

Design of a Meyer Valve. Let us design a Meyer Valve having an eccentricity of 2 inches. Let the eccentricity of the cut-off valve be $2\frac{1}{4}$ inches and the relative travel of the cut-off valve in relation to the main valve be 3 inches. This will make the relative motion of the cut-off valve equivalent to the travel of a plain slide valve with an eccentricity of $1\frac{1}{2}$ inches. Let the outside lap on the main valve be $\frac{3}{4}$ inch, the lead $\frac{1}{32}$ inch, the compression 95 per cent of the stroke, and let the ratio of the length of the crank to connecting rod be six.

In Fig. 56 draw XOY equal to 4 inches, the main valve travel. Lay off YD = 95 per cent of 4 = 3.8 inches, and with a radius of 12 inches, and the center on YX produced draw the arc DH.K. H.K.O is the crank position at compression. C.K.O, the crank position at cut-off, is found in a similar manner. Lay off OI equal to the lap plus the lead, and draw the valve circle for the main valve through I and O with a diameter equal to its eccentricity of 2 inches. To do this take a radius equal to 1 inch, and draw arcs from I and O as centers that shall intersect at B. B is

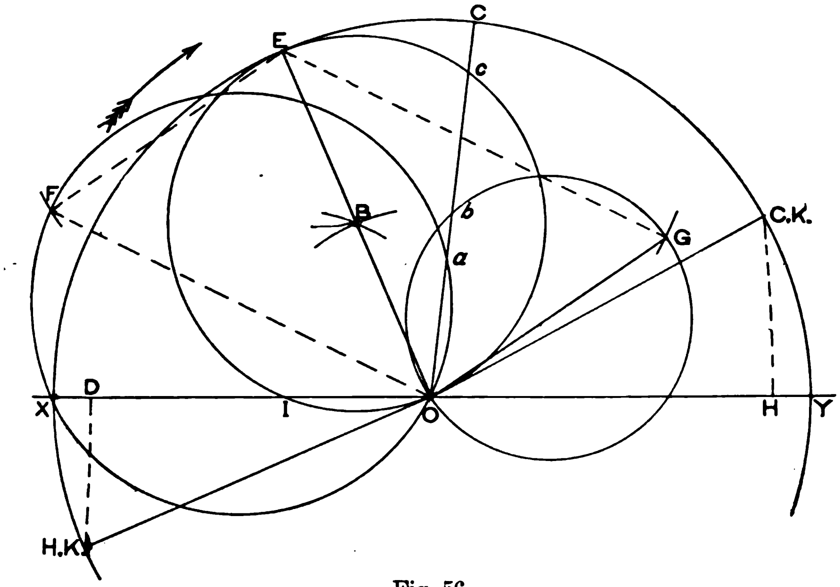

Fig. 56.

the center of the valve circle and OBE is the eccentricity, 2 inches. With E as a center, and with a radius equal to half the relative travel of the cut-off valve (in this case $1\frac{1}{2}$ inches), draw an arc. With O as a center and with a radius equal to $2\frac{1}{4}$ inches, the eccentricity of the cut-off valve, draw another arc intersecting the first one at F. On OF as a diameter construct a valve circle. This valve will represent the absolute motion of the cut-off valve, independent of the motion of the main valve. This circle then will show the displacements of the cut-off valve from the center of the steam chest. With E as a center and with a radius equal to FO draw an arc, and with O as a center and with a radius equal to EF draw another arc intersecting the first at G. On OG as a diameter construct a valve circle. This circle will then represent a travel of the cut-off valve moving on the main valve. That is, it will represent the displacements of the cut-off valve from the center of the main valve. This circle is not, properly speaking, a valve circle and OG is not an eccentricity, but simply represents the relative motion of the two valves. This can be proved by analytical geometry, but an inspection of the figure shows that this must be true.

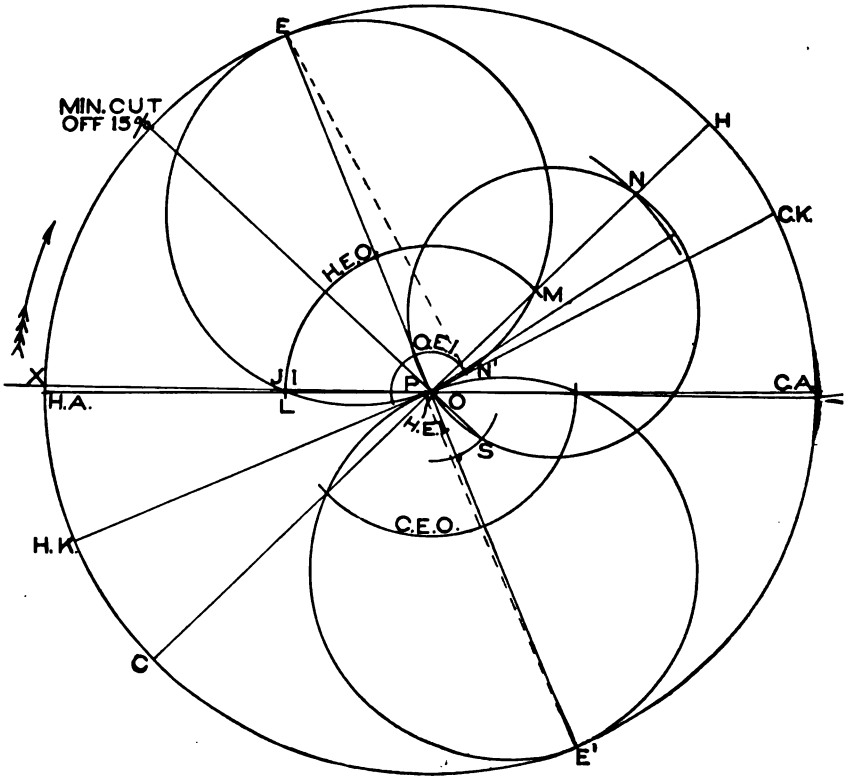

Fig. 57.

Draw the crank line OC at any position, cutting the valve circles at *a* and *b* and *c*. O*a* represents the absolute displacement of the cut-off valve, that is, from the center of the steam chest and O*c* represents the displacement of the main valve. The relative displacement of the cut-off valve, that is, from the center of the main valve, will be the difference between Oc and Oa, since both valves are moving in the same direction. By careful measurement it will be found that $Ob = Oc - Oa$, and any arc as Ob on the auxiliary circle ObG will correctly represent the displacement of the cut-off valve from the center of the main valve at the corresponding crank angle.

Fig. 57 shows the crank angle at head-end compression H.K., and at crank-end compression C.K., the main valve circle, and the auxiliary circle which are transferred from Fig. 56. The construction lines and all lines not essential to the figure are omitted to avoid confusion.

Lay off on Fig. 57, OI equal to the outside lap $\frac{3}{4}$ inch and draw the head-end lap circle H.E.O. It will intersect the valve circle for the main valve at L and M. Through L draw the crank position at admission (head-end) H.A. and the crank position at cut-off through M. This gives the greatest possible cut-off. The cut-off valve may be set to give a much earlier cut-off than this, but of course a later setting would be of no avail for the port would be closed by the main valve at this angle. The crank line OMH cuts the auxiliary circle at N, so that ON ($1\frac{15}{32}$ inches) is the clearance of the cut-off valve. That is, the edge of the cut-off valve must be set $1\frac{15}{32}$ inches from the edge of the main valve port in order to cut-off at this crank angle. The full lines of Fig. 54 show the cut-off valve placed in this position.

The intersection of H.K.O with the lower valve circle, gives the inside lap at the head end of the cylinder. This line comes so nearly tangent to the valve circle that the intersection can be determined only by dropping a perpendicular to H.K.O. from E^1. This cuts the circle at P and $OP = \frac{1}{32}$ inch equals the head-end inside lap, and H.E.I. represents the corresponding lap circle.

The crank-end angle at compression is C.K. which cuts the upper valve circle at N′, giving an inside lap for the crank end of $ON' = \frac{13}{64}$ inch. To make this intersection more apparent the perpendicular can be drawn from E as previously explained.

Suppose that it was required that the minimum cut-off should be 15 per cent. Find the crank position at 15 per cent of the stroke in the same manner as the crank position was found at compression.

Produce this line through O until it cuts the auxiliary circle at Then OS = $\frac{23}{64}$ inch = the required lap for the cut-off valve order to cut-off at 15 per cent of the stroke. The dotted lines Fig. 54 show the cut-off valve drawn in this position.

For a valve of this sort, the cylinder port should be 1½ inche wide and the valve port 1 inch wide. Fig. 54 shows this valve lai out to scale, but as this process is in all respects similar to tha described for laying out a plain slide valve, it will not be describe in detail.

DROP CUT-OFF GEARS.

The ordinary slide valve controls eight different events of the stroke, that is, admission, cut-off, release, and compression for both ends of the cylinder. A change in the setting of a plain slide valve that affects any one event on the crank end, let us say, will also change to a greater or less degree every other event of the stroke, on the head end as well as on the crank end; so that in setting a slide valve the desired position for one event must usually be sacrificed in order to make the others less objectionable.

In order to provide a better distribution of steam than is possible with a single valve, some engines have four valves, two at each end of the cylinder. In horizontal engines, two are placed above the center line of cylinder and two below. The upper are for admission and cut-off, the lower for release and compression. Since each valve controls but two events, a very satisfactory adjustment can be made and the extra complication and cost for large engines are more than overbalanced by the advantages gained, viz.: A very much better distribution of steam, short steam passages and small clearances, separate ports for the admission of hot steam and the exhaust of the same steam after expansion when its temperature has fallen, and finally the possibility of opening and closing the ports very rapidly, thus preventing wire-drawing. The small clearances, short ports and separate admission and exhaust materially reduce the cylinder condensation, and thus effect a large saving in the steam consumption.

When four valves are used for high speeds, the motions of all must be positive, that is, they must be connected directly to some mechanism that either pushes or pulls them through their entire motion, but for speeds up to 100 revolutions or so a disen-

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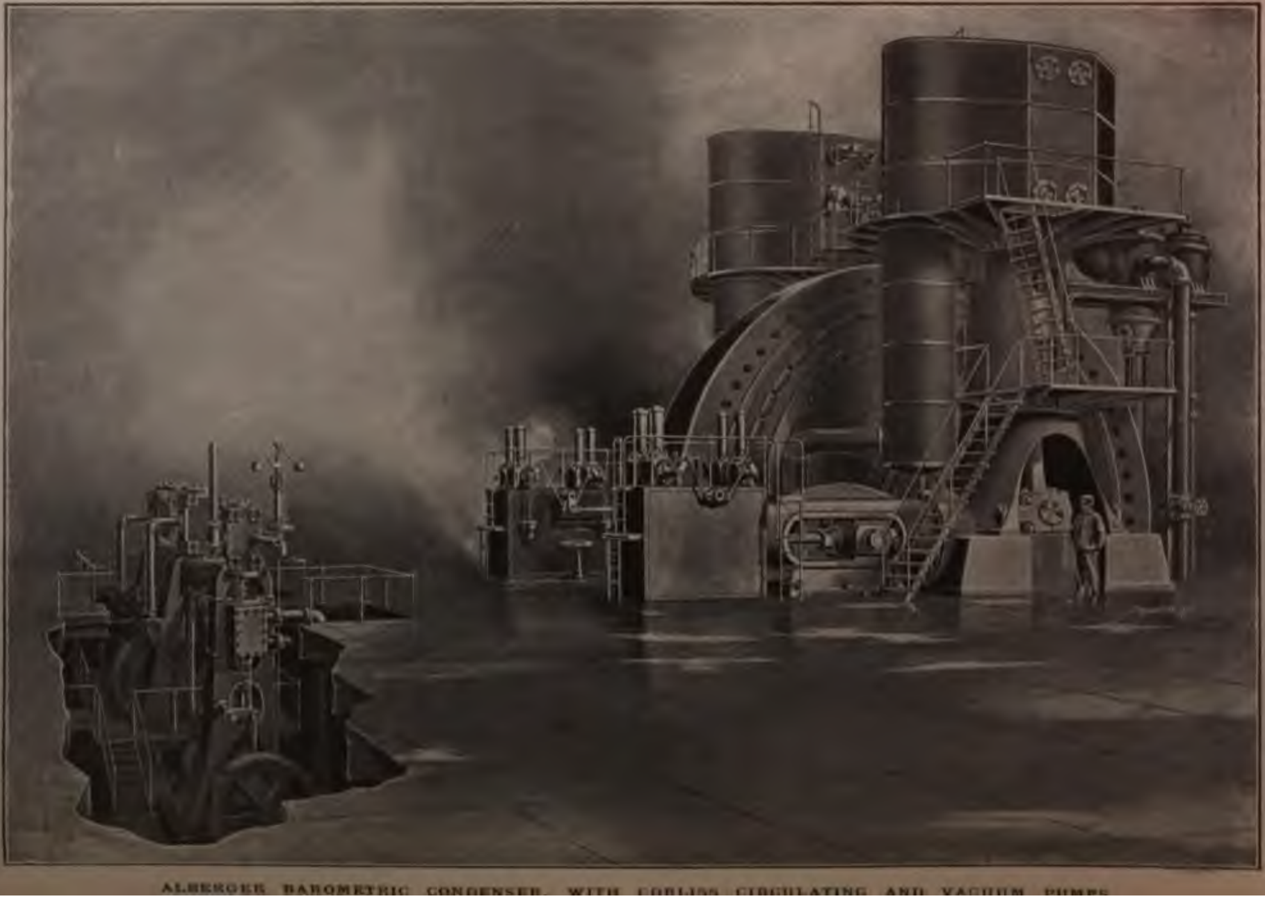

ALBERGER BAROMETRIC CONDENSER, WITH CORLISS CIRCULATING AND VACUUM PUMPS gaging mechanism may be used, and the valves may shut of themselves, either by virtue of their weight or by means of springs or dashpots. The valve is usually opened by means of links or rods, moved by an eccentric, and at the proper point of cut-off the rod is disengaged from the valve which drops shut, hence the term "drop cut-off" gears.

Reynolds-Corliss Gear. The most widely known drop cut-off gear is the Reynolds-Corliss, shown in Figs. 58 and 59; it is often referred to as the *Reynolds hook-releasing* gear. An eccentric on the main shaft gives an oscillating motion to a circular disc called the wrist plate, pivoted at the center of the cylinder. It transmits motion to each of the four valves through adjustable links known as *steam rods* or *exhaust rods*, according to whether they move the admission or exhaust valves.

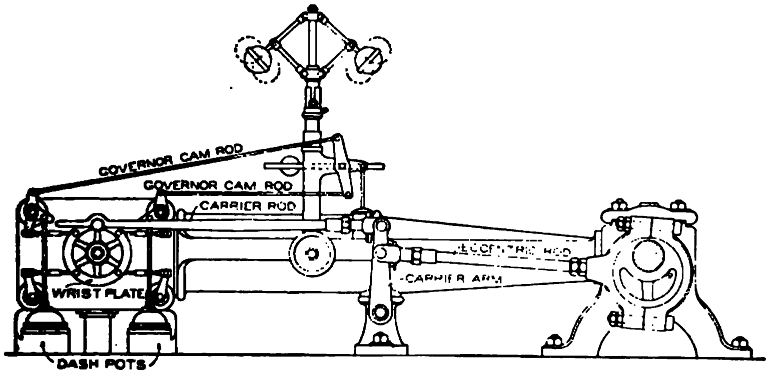

Fig. 58.

The valves which are shown in section in Fig. 60 oscillate on cylindrical seats, and the position of the *rods* is so determined that they give a rapid motion to the valve when opening or closing, and hold it nearly stationary when either opened or closed.

The Reynolds hook is shown in detail in Fig. 59. The *steam arm* is keyed to the valve spindle which passes loosely through a bracket on which the *bell-crank* lever turns, and the spindle is packed to make a steam-tight joint where it enters the cylinder. Motion of the *steam rod* toward the right will turn the *bell-crank* lever and raise the *hook stud*. The *hook* (from which the gear derives its name) pivoted on this stud, has at one end a hard-

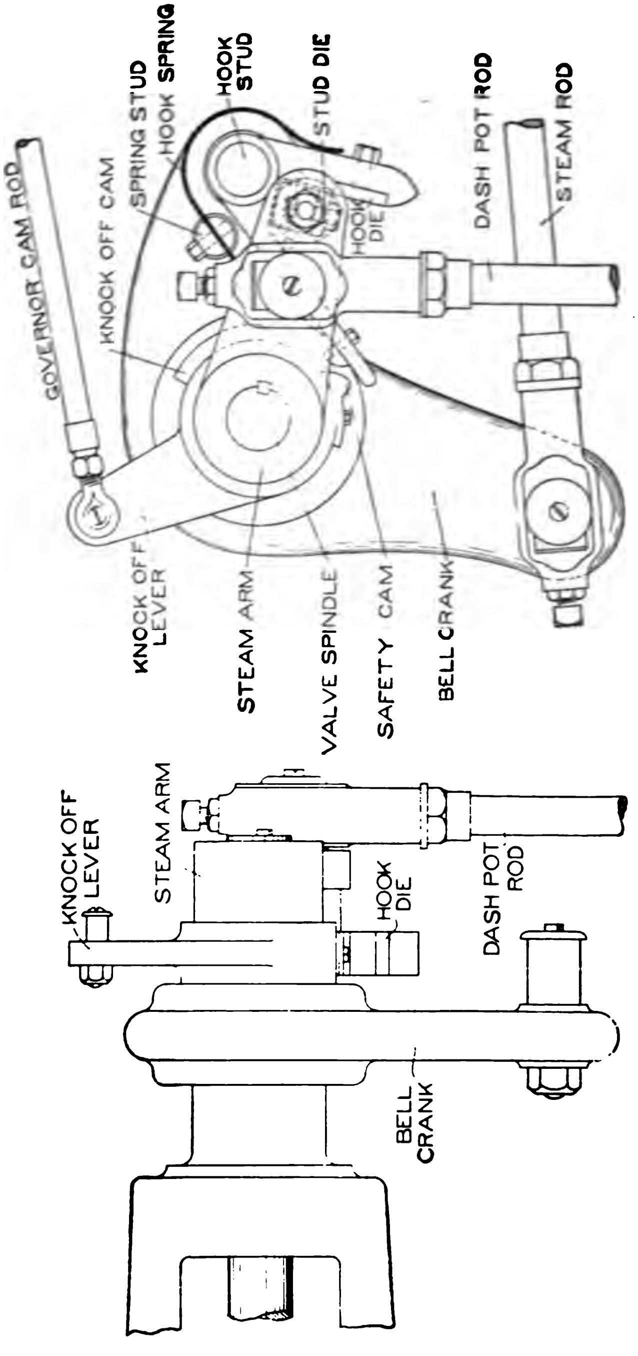
GOVERNOR CAM ROD
KNOCK OFF CAM
SPRING STUD
HOOK SPRING
HOOK STUD
STUD DIE
HOOK DIE
DASH POT ROD
STEAM ROD
KNOCK OFF LEVER
STEAM ARM
VALVE SPINDLE
SAFETY CAM
BELL CRANK
KNOCK OFF LEVER
STEAM ARM
HOOK DIE
DASH POT ROD
BELL CRANK

Fig. 50.

d steel die with sharp, square edges, and at the other end, mall steel block with a rounded face. As the hook rises, the *k die* engages the *stud die* which is fastened to the *steam arm*, l one end of the steam arm is thus raised. This turns the ve in its seat and admits steam. As the *hook* continues to e, its stud moves in an arc above the valve spindle, and the nd-faced block at its left-hand end strikes the *knock-off cam* ich causes the *hook* to turn about its stud and disengage the *k die* from the *stud die*. In raising the *steam arm*, the *dash-* *rod* also is raised and a partial vacuum is created in the dash- . As soon, therefore, as the dies become disengaged, the *hpot rod* quickly drops under the force of this vacuum, thus ning the steam arm and closing the valve. The striking of the -hand end of the hook against the knock-off cam determines point of cut-off, by releasing the valve at that instant.

This cam is a part of the *knock-off lever* to which the *governor* *rod* is fastened. Any action of the governor which would se the *cam rod* to move toward the right would cause this *ck-off lever* to turn on its axis, the steam arm, and conse- ntly lower the position of the *knock-off cam*. This would se an earlier contact between the cam and end of hook, and sequently an earlier cut-off. By lengthening or shortening *governor cam rod*, the point of cut-off can be adjusted to suit engine load without changing the speed.

There is a limit to this adjustment, for it can be shown that orliss gear operated by a single eccentric cannot be arranged to off later than half stroke. Suppose the eccentric is set just 90° ad of the crank. Then it will reach its extreme position just he piston gets to half stroke. If by that time the *hook* which rising and opening the admission valve, has not yet struck *knock-off cam*, it cannot strike it at all, for any further motion cause the hook to descend to its original position, that is its tion at the beginning of the stroke; the hook will not disen- e from the steam arm *stud* at all and the bell crank will rn, closing the valve in the same manner in which it opened Cut-off will then take place near the end of the stroke, but it not be the sharp cut-off produced by the sudden *drop* when dies are disengaged.

If the eccentric were set less than 90° ahead of the crank, the cut-off could be arranged to occur later than half stroke, but this is decidedly impracticable, for with such a position of the eccentric the action of the valves at release and compression is spoiled. When it is necessary to cut-off later than half-stroke, as sometimes happens on low-pressure cylinders of compound engines, it may be arranged for by means of two eccentrics, one set more than 90° ahead of the crank to operate the exhaust valves, and one less than 90° ahead to operate the admission valves.

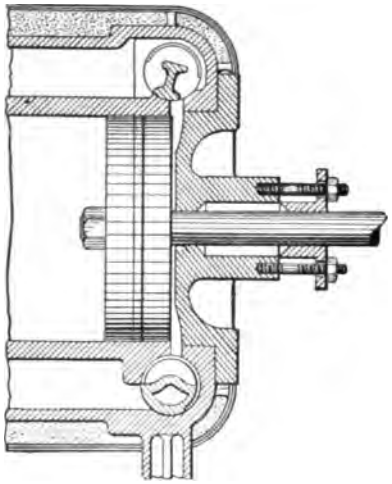

Fig. 60.

The safety cam shown in Fig. 59 is an important part of a Corliss gear. If for any reason the engine governor should fail to act, due, for instance, to the breaking of its driving belt, the governor would drop to its lowest position, supply more steam to the engine and allow it to run away. The *safety cam* prevents this by moving so far to the right that it strikes the hook when it descends to pick up the steam arm. The hook is consequently turned toward the right and then lifted without engaging the stud die; the valve consequently remains closed and the engine stops.

Brown Releasing Gear. In addition to the *Reynolds hook*, several other devices are in use for opening and releasing Corliss admission valves. Among them the Brown releasing gear shown in Fig. 61 may be noted. The steam rod and dashpot rod are arranged much the same as in the Reynolds gear. The governor cam rod operates a plate cam having a curved slot so shaped that it takes the place of both the knock-off and the safety cam of Fig. 59. The steam arm is keyed to the valve spindle and carries at its lower end a steel die which is free to slip up and down a small amount. The part of this gear corresponding to the Reynolds bell crank becomes a straight rocker pivoted at its middle;

and the part corresponding to the Reynolds hook has at one end a die which engages the die of the steam arm, and at its other end a roller running in the curved cam slot. This hook is really a bell-crank lever with arms that are not in the same plane. The bearing on which it turns is carried on the lower end of the rocker, and therefore is equivalent to a movable pivot similar to the hook stud of the Reynolds gear.

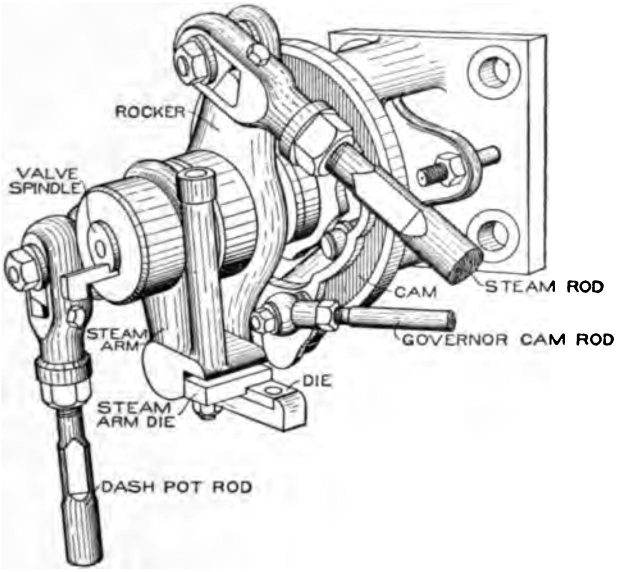

Fig. 61.

In the position shown the dies are engaged. Motion of the steam rod toward the right will move the lower end of the rocker toward the left, and consequently turn the valve spindle in a right-handed direction. This will open the valve and at the same time raise the dashpot rod. Meanwhile, the roller is moving toward the left in a circular part of the cam slot, the center of which is at the center of the valve spindle. This causes the steam arm and the bell-crank lever, which has the roller at one end, to move in such a way that there is no relative motion between them. As soon, however, as the roller comes to the point where it is forced to move out of this circular path and move farther from the valve spindle, both arms of the bell-crank lever are turned downward, the dies become disengaged, and the dashpot closes the valve. The slight up-and-down motion of the steam-arm die allows it to rise while the hook die passes underneath when returning to re-engage for the next stroke. The makers claim that this gear permits a much higher speed than is possible with other Corliss gears.

Greene Gear. Another well-known drop cut-off gear is the Greene, shown in Fig. 62. The valves are of the gridiron type, sliding on horizontal seats, the admission valves parallel to, and the exhaust valves at right angles to the axis of the cylinder and just below it. AA are rock shafts turning in fixed bearings.

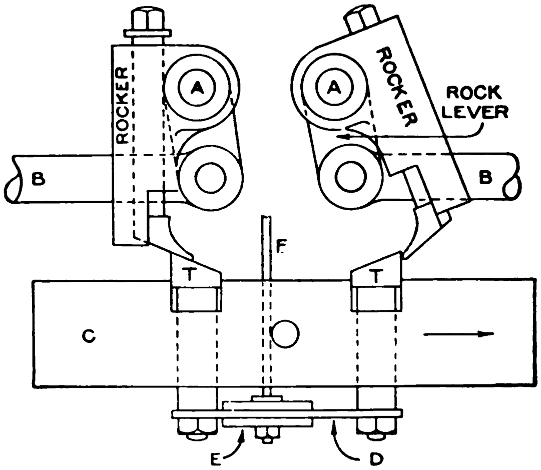

Fig. 62.

BB are the admission valve stems. C is a slide bar, receiving a reciprocating motion from an eccentric. TT are tappets connected to the slide bar. They move to and fro with the slide bar and can also move independently up and down. They are made fast at their lower end to the gauge plate D which slides through the guide E. The guide E is made fast to the governor rod F and through this means can be raised or lowered, thus regulating the height of the tappets.

As the slide bar moves toward the right, the right-hand tappet is brought into contact with the toe of the rocker, causing it to turn on its bearings and move the rock lever and the valve stem B toward the right, thus opening the admission valve. Since the tappet moves in a horizontal direction while the toe of the rocker moves in an arc, it will be seen at once that they will soon become disengaged and release the valve which is at once closed by a dashpot not shown in the figure. If the governor raises the tappets, cut-off will be later. A nut at the bottom of the governor rod allows a proper adjustment of the guide and gauge plate. As the slide plate C moves toward the right, the left-hand tappet comes in contact with the heel of the left-hand rocker, both being beveled, it rises in its socket allowing the tappet to pass under. It then falls by its own weight and is ready to engage the tappet on its return and open the valve. In this gear the disengagement of the valve throws no load whatever on the governor which is a distinct advantage over the Corliss gear. The action of the exhaust valves is not shown in the cut.

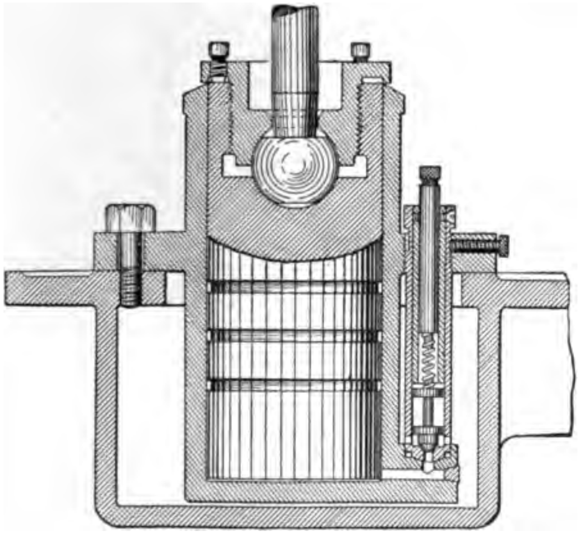

Fig. 63.

The Sulzer Gear is a drop cut-off widely used in Europe. The valves are of the poppet type, lifting straight from conical seats, so that there is no friction. They are usually placed vertically above and below the cylinder axis and are operated by eccentrics from a shaft geared to the main shaft. The admission valves are lifted from their seats by suitable levers, then released by a device equivalent in action to the Reynolds hook and are quickly closed by the action of springs.

The exhaust valves of all drop cut-off gears are comparatively simple in their operation and both in opening and closing they are moved by the direct action of the exhaust rods.

A common form of vacuum dashpot for closing admission valves is shown in Fig. 63. The rod coming down from the steam arm makes a ball-and-socket joint with the dashpot piston. The dashpot is often let down into the engine frame as shown. When lifted, the piston produces a partial vacuum underneath it so that it tends to drop quickly as soon as the valve gear is released. On some of the largest modern engines where the valves are very heavy, steam-loaded dashpots are used; that is, the dashpot piston has steam pressure on one side, and an air cushion on the other prevents it from striking the bottom of the dashpot.

Corliss Valve Setting. The setting of a Corliss valve gear is a much longer process than the setting of a plain slide valve, but is nevertheless a comparatively simple matter, for the various adjustments are nearly all independent of one another. In gears like that shown in Fig. 58 the length of both the eccentric rod and carrier rod are unusually adjustable, and the former should be of such length that the carrier arm swings equal distances on each side of a vertical line through its pivot, and the carrier rod should be adjusted until the wrist plate oscillates symmetrically about a vertical line through its pivot Nearly all Corliss engines have one mark on the wrist plate hub and three on the wrist plate stand, as shown in Fig. 64, and the wrist plate should swing so that A, the mark it carries, moves from C to D, but not beyond either one. When A is in line with B, the wrist plate is in mid-position. The valves are then not in their exact mid-position, but it is customary to regard them as being in mid-position, and to speak of the lap as the amount the valve covers the port when the wrist plate is in mid-position.

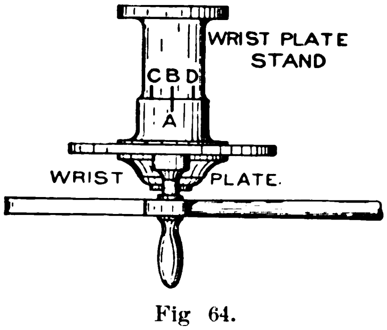

Fig 64.

To set the valves, remove the bonnets or covers of the valve chambers on the side opposite the gear. The ends of the valves are circular, but inside their cross-section is as shown in Fig. 65. On the end, in line with the finished edge of the valve, and on the seat in line with the edge of the steam port, are marks as shown in Fig. 65. When these marks coincide, the valve is either just opening or just closing, and when in any other position, the amount of opening or the amount by which the port is closed is shown directly by the distance between the marks. Block the wrist plate in mid-position, hook up the admission valves and adjust the length of the steam rods by means of the right and left threads provided for the purpose, until the ports are covered by the amount of lap indicated in the following table opposite the given size of engine.

Dia. of Cyl. in inches.	Steam Lap in inches.	Exhaust Clearance in inches.
12	$\frac{1}{4}$	$\frac{1}{32}$
14 to 16	$\frac{5}{16}$	$\frac{1}{32}$
18 to 22	$\frac{3}{8}$	$\frac{1}{16}$
24 to 28	$\frac{7}{16}$	$\frac{3}{32}$
30 to 36	$\frac{1}{2}$	$\frac{1}{8}$
36 to 42	$\frac{5}{8}$	$\frac{5}{32}$

Next adjust the exhaust rods until the exhaust ports are open an amount equal to the clearance given in the above table. Set the engine on its head-end dead point, hook the carrier rod onto the wrist plate and in the direction in which the engine is to run, turn the eccentric enough to open the head-end admission valve by a proper amount of lead; then the eccentric will be 90° plus the angular advance ahead of the crank. The proper amount of lead will depend upon both the design of the gear and the speed at which the engine is to run; and may vary from $\frac{1}{32}''$ for small engines to as much as $\frac{5}{32}''$ or $\frac{3}{16}''$ for large and higher-

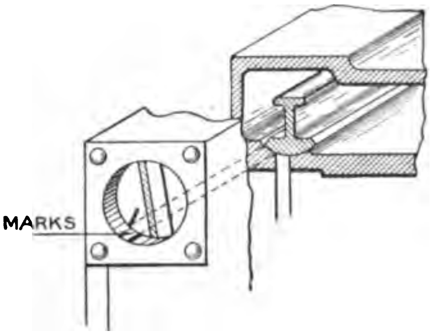

Fig. 65.

speed engines. When the proper amount of lead has been obtained, fasten the eccentric on the shaft by means of the set screw and make sure by trial that the wrist plate moves to its extremes of travel. The dashpot rods must be adjusted so that when the dashpot piston is at its lowest position, the hooks (see Fig. 59) descend just far enough for the hook dies to snap over the stud dies with about $\frac{1}{32}$" to $\frac{1}{16}$" to spare, depending on the size of the gear.

To adjust and equalize the cut-off, lift the governor to about the position that it will occupy when running at normal speed, and put a block under the collar to hold it in this position. First set the double lever at the right of the governor cam rods so that it makes approximately equal angles with each rod, and then turn the engine over by hand until the piston has moved to the desired point of cut-off. Adjust the proper cam rod until the knock-off cam strikes the hook and allows the valve to close, then turn the engine to the point of cut-off on the other stroke and adjust the other cam rod in a similar manner. Now set the governor in the lowest position to which it could fall if there were no load on the engine, and set the safety cams so that in this position the hook cannot open the valve. A latch is provided on which the governor can be supported slightly above its lowest position so that the valves can be opened by the hook when starting the engine. As soon as the engine speeds up this latch must be moved aside, so that if the governor fails to act, it can drop to its lowest point, otherwise this latch would hold it just high enough so that the safety cams could not act.

When Corliss gears are set as here described, the position of the eccentric may not be quite right, due to an incorrect estimate of the amount of lead required. The error is likely to produce faulty release and compression as well as poor admission, but it cannot be very serious, and the engine will turn over with its own steam, so that indicator diagrams may be taken. The final adjustments can then be determined from an examination of the diagrams.

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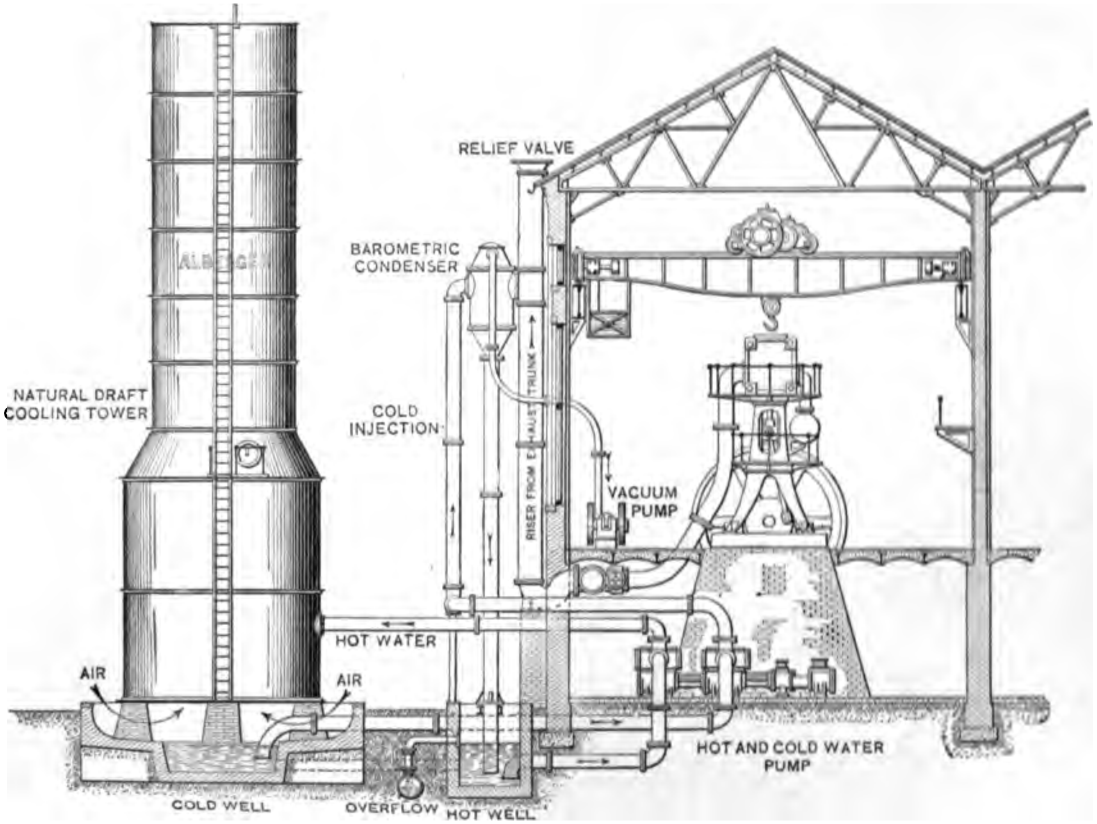


BAROMETRIC CONDENSER AND NATURAL DRAFT COOLING TOWER.

Alberger Condenser Co.

THE STEAM ENGINE.

PART II.

ACTION OF HEAT.

There are several types of heat engines, such as the steam engine, the gas engine, the hot-air engine, etc., each one of which derives its motive power from the heat contained in steam, gas, oil, hot air, or some other substance. Heat is imparted to these substances either by the combustion of fuel in a generator entirely separate from the engine, or by the combustion of a gaseous substance in the cylinder of the engine itself. In the case of the hot-air engine the heat is produced by a fire immediately beneath the engine.

Steam and hot gases have a tendency to expand, because of the heat they contain, thus producing a pressure in all directions. This pressure causes the piston of the engine to move, which allows the gas to expand. As the gas expands, it gives up heat, which is converted into useful work.

We shall now discuss the fundamental principles of the action of heat, and the behavior of gases and vapors, with special reference to the properties of steam and its action in the cylinder of the engine.

If a piece of iron or some other substance is placed in a fire, it becomes hotter than it was before, because heat from the fire has passed into it. If this hot substance is plunged into cold water, or is allowed to remain in a cool place, heat will pass from it, and we say that it becomes cold.

There have been many theories as to what heat is. The accepted theory of to-day is that heat is the result of motion, or we may say it is a form of kinetic energy. Heat is produced, not by the motion of the substance itself (for the hot body may be at rest), but by a rapid vibration of the minute individual particles that make up the body. These minute particles are called molecules. The faster the molecules vibrate, the greater will be their kinetic energy and the hotter the substance will become. The hottest bodies are those that have the greatest energy of molecular vibration. A hot body can transfer its energy of vibration to another, which in turn becomes hotter than it was before, while the first body loses a part of its energy of vibration and becomes cooler. The terms hot and cold are only comparative terms; one body is hot because it contains a greater degree of heat than another; the other is cold because it contains a less degree of heat than the first. Cold, then, is simply a low degree of heat.

By temperature is meant simply the thermal condition of a body with reference to its capability of transferring heat to other bodies. If two bodies are placed in contact and the first gives more heat to the second than it receives, we say that No. 1 is hotter than No. 2. If the first receives more heat than it gives, No. 2 is hotter than No. 1. If both bodies give and receive the same amount of heat, they are of the same temperature.

According to our theory, it is evident that temperature depends upon the energy of molecular vibration. If the temperature rises, it means that the molecular vibration, and consequently the energy, increases; if the temperature falls, the energy of molecular vibration decreases. Evidently a point must finally be reached when this energy of vibration is zero and the molecules are at rest. At this temperature there is no heat and we call it the absolute zero. It is evident that this zero is very much below the zero of the ordinary scale.

In order to determine just how hot a body is, we must compare its temperature with that of some substance whose degree of heat we know. It would be impossible to keep several bodies at different degrees of heat for comparison, so we must resort to some other means. A simple method is to use some substance whose volume changes a definite amount for a definite change in temperature and always has the same volume for the same temperature. Mercury and alcohol are suitable substances and may be placed in a glass bulb, to which is connected a glass tube of small bore. All the air is drawn out of the tube, and the end is sealed so that the thermometric substance can expand or contract in a vacuum. The tube having been sealed, the bulb is placed in melting ice and the height of the mercury in the tube noted. It is then placed in steam (or boiling water) at atmospheric pressure and the height of the column again noted. On the Fahrenheit scale the melting point is called 32°, and the boiling point 212°, and the intervening space is divided into 180 equal parts. In the Centigrade scale the melting point is called 0 and the boiling point 100°; there are 100 equal intervals between them. Thus we see that $180° \text{ F} = 100° \text{ C}$, or $1° \text{ C} = 1.8° \text{ F}$.

Example: What is the temperature of 50° C on the F scale?

$$50° \text{ C} = 50 \times 1.8 = 90° \text{ F above the melting point}$$
$$\text{or } 90 + 32 = 122° \text{ F above zero.}$$

In order to compare temperatures, we place the thermometer in contact with the substance whose degree of heat we wish to know and then observe the height of the liquid column in the thermometer. The height of this column depends upon the expansion of the thermometric substance and indicates the intensity of heat, or the temperature as we commonly call it. We use a thermometer to measure the *intensity* of heat, but not the *quantity* of heat.

For measuring the intensity of heat, the degree is the unit; for measuring the quantity of heat we have another unit, which is the amount of heat necessary to raise one pound of water from 61° F to 62° F. This is called the British thermal unit (B. T. U.). To raise one pound of water from 60° F to 62° F, or to raise two pounds from 60° F to 61° F, will require 2 B. T. U.

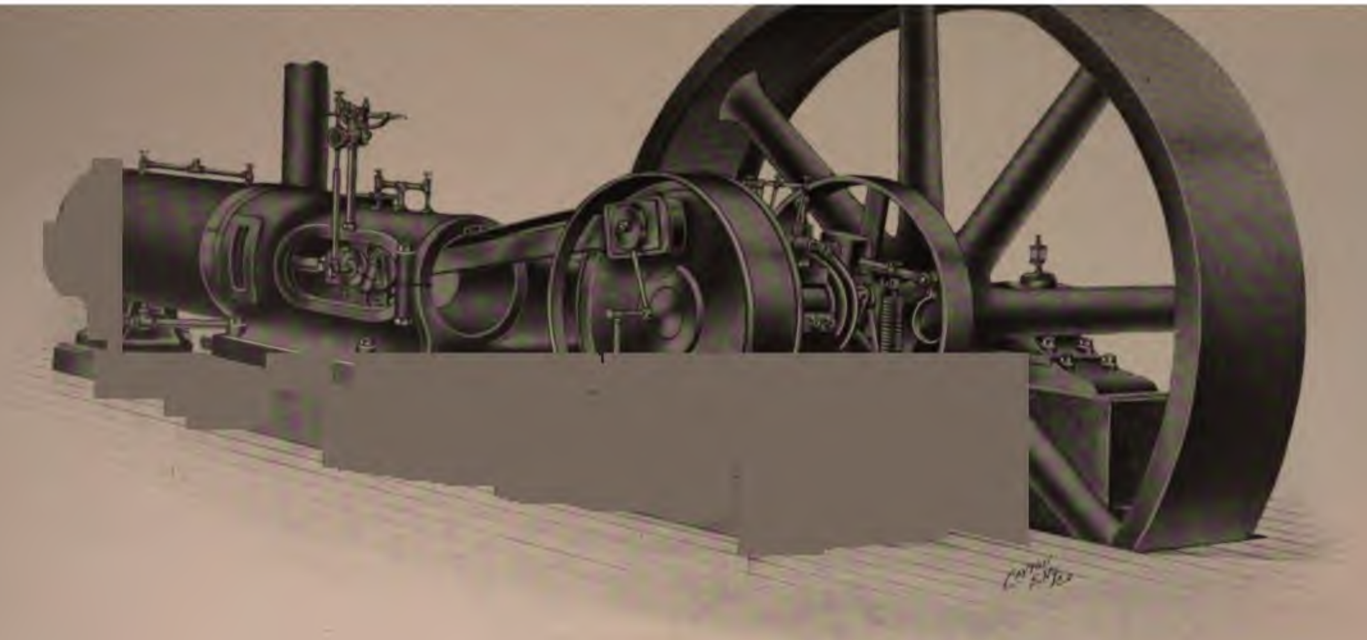
Suppose we have a small bar of iron heated to a white heat; its temperature will be high, but it will contain a relatively small quantity of heat, that is, it will not require a great many B. T. U. to raise its temperature to this high point. But suppose this same number of heat units were transferred to a ton of iron; the temperature would scarcely be changed, for the infinitely greater number of particles would have a correspondingly less number of vibrations.

Experience teaches us that when a rifle-ball strikes a target it stops, and that the energy which it possesses by virtue of its bodily motion is suddenly transformed into energy of molecular motion. Energy is indestructible; the molecular motion of the impinging body is at once increased and heat is developed. In general, whenever moving bodies are brought to rest, either suddenly as by impact or gradually as by friction, the kinetic energy of the moving mass is transformed into molecular kinetic energy, and we say that the bodies become heated.

We have now seen that we can transform energy of motion into heat. Also by means of suitable apparatus we can transform heat into energy of motion. Heat is the lowest form of energy, and while it is comparatively easy to transform other forms into heat, it is not as easy to change heat into the higher forms of energy. The principle of the transformation is simple enough, but if we are to have an efficient engine, we must be able to extract practically all of the heat from the working substance. This, however, is impossible, because the ordinary ranges of temperature used in practice are so far removed from the absolute zero of temperature, that with the most perfect machines we can at best recover but a fraction of the heat; the rest passing out of the engine.

The transformation is accomplished by means of a working substance which passes from the temperature of a heat generator into a refrigerator; the heat given up during the change is transformed into work. By the term refrigerator we mean the low temperature of the working substance at exhaust. The greater the temperature of the source of heat, or the lower the temperature of the refrigerator, the greater will be the amount of heat that can be abstracted and converted into useful work. If the temperature of the refrigerator could be reduced to the absolute zero, all the heat would be removed from the working substance, and the only loss would be that due to mechanical imperfections of the engine; but since the absolute zero is 461° below the zero on the Fahrenheit scale, or 493° below the freezing point, we must at best allow a relatively large amount of heat to pass out of the engine into the refrigerator without having done any work at all.

The unit of work is the foot-pound; that is, the work done in raising one pound one foot. By means of careful experiments it has been determined that for every 778 foot-pounds of work transformed into heat there is developed one B. T. U. This value 778 is known as the Mechanical Equivalent of heat. It means that one B. T. U. and 778 foot-pounds of work are mutually interchangeable.



HEAVY DUTY, MEDIUM SPEED ENGINE.
Buckeye Engine Company.

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EXPANSION OF GASES.

A perfect gas, strictly speaking, is one that cannot be liquefied ; ordinarily, however, we apply the term to those gases that can be liquefied only with great difficulty, that is, under extreme pressure and a great reduction of temperature. For every perfect gas there is a definite relation between the pressure and volume, and what is known as Boyle's Law has been found to hold true, *viz.:* The pressure of a perfect gas at constant temperature varies inversely as the volume; that is, if the temperature remains constant the pressure becomes less as fast as the volume becomes greater, or conversely the volume becomes less as fast as the pressure becomes greater. Now if the pressure becomes twice as great, the volume becomes half as large, and if the volume becomes three times as great, the pressure will be only one-third as much. Hence we see that however the pressure may vary, the volume will change in such a way that the pressure multiplied by the volume will always be constant, provided the temperature remains the same.

This simple law is expressed thus:

$$P \times V = C$$

In which

$P =$ pressure in pounds per square inch (absolute)

$V =$ volume in cubic feet

$C =$ a constant which has different values for different gases.

Gases that are not easily liquefied, such as hydrogen, oxygen and air, follow this law fairly well, but those that are easily liquefied, such as steam and ammonia, do not follow it at all.

The value of C is not the same for all gases, and as no gas is, strictly speaking, a perfect gas, its value varies slightly with the temperature. For air, which is nearly a perfect gas, its value is 182.08 at 32° F.

Example.—What is the absolute pressure per square inch of one pound of air if the temperature is 32° F, and the volume 4.129 cubic feet?

$$V = 4.129$$

$$P \times V = C \quad \text{or} \quad P = \frac{C}{V} = \frac{182.08}{4.129}$$

$P = 44.1$ pounds per square inch, absolute.

Let us discuss this law by means of a diagram. Fig. 1 is drawn with O Y and O X at right angles to each other; pressures are measured to any convenient scale on O Y and volumes on O X. O C represents 12.387 cubic feet and O D represents 14.7 pounds, since P × V = 182.08. O F represents 24.774 cubic feet and O G 7.35 pounds. In like manner O L represents 6.19 cubic feet and O N 29.41 pounds. Then if we draw perpendiculars to O X at L, C and F, and perpendiculars to O Y at N, D and G, they will meet in the points M, E and H. The curve A B is drawn through these points. Then for any pressure we can find the corresponding volume or *vice versa*. The area of the rectangle O D E C equals that of the rectangle O G H F and also that of the rectangle O N M L. That this is so is readily seen from the fact that the product of the pressure and its corresponding volume is constant.

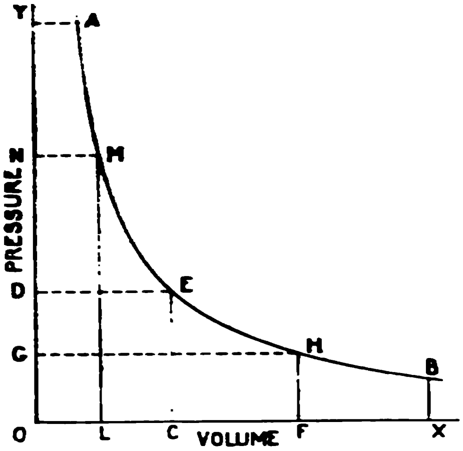


Fig. 1.

If we plot a curve using this equation we will get a rectangular hyperbola, as shown in "Steam Engine Indicators." This curve is called an isothermal curve or curve of equal temperatures.

If the volume of a perfect gas remains constant, the pressure will vary as the temperature. Or, if the pressure remains constant, the volume will vary as the temperature. This is known as the Law of Charles. From this we see that, as the temperature decreases, either the pressure or the volume will decrease a proportionate amount, and this must continue as long as there is any heat in the gas; finally a low temperature (the absolute zero) will be reached, where there is no more heat, and consequently either the pressure or the volume must be zero, provided this law holds true at such a very low temperature.

The law states that any change of pressure or volume is proportional to the change in temperature, that is, the new volume is equal to the first volume plus or minus some fractional part of this volume, called the coefficient of expansion, multiplied by the change in temperature. We may express the formula thus:

$$P_1 V_1 = P V + (P V) k \times t$$
$$= P V (1 + k t).$$

Where P_1 and V_1 are the new pressure and volume
P and V are the first pressure and volume
k is the coefficient of expansion for the gas
t is the change in temperature.

From Boyle's Law we know that $P V = C$; hence we may write the above equation:

$$P_1 V_1 = C (1 + k t)$$
$$= K C \left(\frac{1}{k} + t\right)$$

From careful experiments on the expansion of air, Regnault determined the value of the coefficient k to be .003654 for Centigrade units. This value is constant between freezing point, 0, and boiling point, 100°. Substituting this value of k in our last equation, we have:

$$P_1 V_1 = C \left(\frac{1}{.003654} + t\right)$$
$$= C (273.7 + t).$$

Now if we should make the change of temperature, t, equal to $-273.7°$ we should have:

$$P_1 V_1 = C(273.7 - 273.7) = 0$$
$$P_1 V_1 = 0.$$

Therefore we must have reached the absolute zero at $-273.7°$ below the freezing point, because here $P_1 \times V_1$ has reached 0, as we have previously seen must be the case at the absolute zero. $273.7 \times 1.8 = 492.7°$ for the F scale. Therefore the absolute zero is $492.7° - 32° = 460.7°$ below zero on the Fahrenheit scale. For ordinary work 461 will be sufficiently accurate.

If we let P = absolute pressure in pounds per square inch
V = volume of one pound in cubic feet
T = absolute temperature on Fahrenheit scale,

Then, $$P V = C T.$$

C equals .3693 for air.

Example.—If one pound of air occupies 16.606 cubic feet at a pressure of 14.7 pounds per square inch, what is its temperature?

$$P \times V = C \times T$$

$$14.7 \times 16.606 = .3693 \times T$$

$$T = \frac{244.108}{.3693}$$

$$T = 661.$$

This value 661 is absolute temperature, and to find the Fahrenheit temperature 461 must be subtracted from it. Thus, 661° — 461° = 200° F.

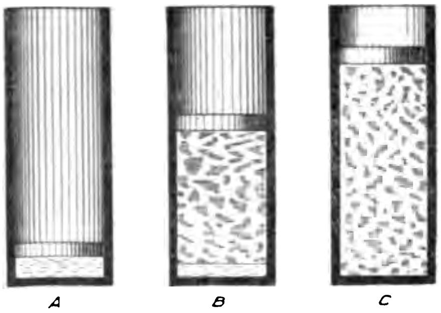


Fig. 2.

Saturated Vapor. The process of converting a liquid into a vapor is known as vaporization; the product thus formed is readily condensed and therefore does not follow the laws of perfect gases at all. A dry saturated vapor is one that has just enough heat in it to keep it in the form of a vapor; if we add more heat it becomes superheated. A superheated vapor may lose a part of its heat without condensation; a saturated vapor cannot. When a saturated vapor loses a part of its heat some of it will condense and we say that the vapor is wet.

Steam is simply the vapor from water and we shall confine our discussion to this alone. Suppose we have a vertical cylinder, as shown in Fig. 2, fitted with a light piston free to move up and down, yet so constructed that it may be loaded at will. Suppose that there is one pound of water at a temperature of 32° F in the bottom of this cylinder, and that the piston rests upon its surface. Now, if we apply heat by means of a gas flame or fire, we shall notice the following effects:

First.—The temperature of the water will gradually rise until it reaches the temperature at which steam is formed. This temperature will depend upon the pressure, or the load on the piston. If the piston is very light, we may neglect its weight and consider that there is simply the atmospheric pressure of 14.7 pounds per square inch acting on the water surface. At this pressure steam will begin to form at 212° F.

Second.— As soon as 212° F is reached, steam will begin to form and the piston will steadily rise, but no matter how hot the fire may be, the temperature of both water and steam will remain at 212° until all the water is evaporated. We had one pound of water at 32° F and at 14.7 pounds absolute pressure, and found that steam formed at a temperature of 212° F and remained at that temperature. We added 180.9 B. T. U., the heat of the liquid, to bring the water from 32° to the boiling point. To convert water at 212° into steam at 212°, we added 965.7 B. T. U. more. This quantity, known as the latent heat, or heat of vaporization, makes the total heat 1,146.6 B. T. U. If we should measure the volume carefully after all the water was evaporated, we should find that there was just 26.36 cubic feet of dry saturated steam. We had one pound of water, and therefore must have one pound of steam, for none of it could escape; hence one cubic foot will weigh $\frac{1}{26.36} = 0.03794$ pounds, which is known as the density of steam at 14.7 pounds absolute pressure or 212° F. In the Table of Properties of Saturated Steam (see page 14) all these quantities are found in the order given and at the pressure of 14.7 pounds above vacuum.

Suppose now we place a weight of 85.3 pounds on the piston. The pressure is 85.3 pounds plus 14 7 pounds, or 100 pounds absolute. We shall now find that no steam will form until a temperature of 327.58° is reached. Starting with water at 32°, it will be necessary to add 297.9 B. T. U. before a temperature of 327.58° is reached, and also we must add 884.0 B. T. U. more to vaporize it, making a total heat of 1,181.9 B. T. U. Under this greater pressure the steam occupies a volume of only 4.403 cubic feet, or one cubic foot of it weighs $\frac{1}{4.403} = 0.2271$ pound.

Of course it would be impossible to determine all these different quantities by actual experiment, and at all pressures varying from vacuum to the high pressures, used in water-tube boilers.

Fortunately we are able to compute them all from equations which have been carefully determined by experiment. If saturated steam were a perfect gas, we could easily calculate all the relations of pressure, volume and temperature from the equation $PV = CT$, but steam is so far removed from the state of a perfect gas that these relations do not hold, and the true equations become very complex. The following equation proposed by Rankin is one of the simplest, and gives fairly good results:

$$\log_{10} P = A - \frac{B}{T} - \frac{C}{T^2}$$

in which P = pressure in pounds per square inch above vacuum.
A = 6.1007
B = 2,732
C = 396,945
T = absolute temperature in Fahrenheit degrees.

By the aid of such equations as this, all the different quantities found in the steam tables may be calculated. These equations are based on careful experiments and give very satisfactory results.

Steam Tables. We have already seen that any change in the temperature of saturated steam produces a change of pressure, and that every change of pressure corresponds to a certain change in temperature. There are several properties of saturated steam that depend upon the temperature and pressure; and the values of all these different properties when arranged for all temperatures and pressures are called Steam Tables. The following are the principal items that are found in the tables:

1. The absolute pressure in pounds per square inch; it is equal to the gage pressure plus the atmospheric pressure of 14.7 pounds.

2. The temperature of the steam, or boiling water, at the corresponding pressure.

3. The heat of the liquid; or the number of B. T. U. necessary to raise one pound of water from 32° F to the boiling point corresponding to the given pressure.

4. The heat of vaporization, or the latent heat; this is the number of B. T. U. necessary to change one pound of water, at the boiling point, into dry saturated steam at the same temperature and pressure.

5. The total heat or the number of B. T. U. necessary to change one pound of water from 32° F into steam at the given temperature or pressure. The total heat is evidently equal to the sum of the heat of the liquid and the heat of vaporization.

6. The density of the steam; that is, the weight in pounds of one cubic foot of steam at the given temperature or pressure.

7. The specific volume; or volume in cubic feet of one pound of steam at the required temperature or pressure. Evidently the specific volume is equal to $\frac{1}{\text{density}}$.

All these properties have been calculated by means of various formulas which have been deduced from the results of actual experiment. There are several formulas for the temperatures and pressures of steam; as some computers have used one and some another, there is likely to be a slight discrepancy between the tables computed by different authors. Rankin's formula, already given, is the simplest, but is not generally considered to be quite as accurate as some of the later ones; it has probably been used, however, more than any other.

The total heat may be calculated by means of the formula

$$H = 1{,}091.7 + 0.305\,(t - 32) \qquad [1]$$

in which H = total heat

t = temperature in degrees Fahrenheit.

The heat of the liquid is equal to q.

$$q = t + 0.00002\,t^2 + 0.0000003\,t^3 \qquad [2]$$

in which t = temperature in degrees Centigrade.

These constants are for use in the Centigrade system only. To calculate the heat of the liquid for any Fahrenheit temperature it is necessary to change the Fahrenheit into equivalent Centigrade degrees and then substitute in the above formula.

The formula used for the calculation of specific volume is too complex for consideration here, but the relation of pressure and volume may be approximately expressed by means of an equation of the form $P V^n = C$, in which n is an exponent, C a constant, P the absolute pressure, and V the specific volume; n is usually taken to be $\frac{17}{16}$, and C to be 475.

On pages 14 and 15 are given tables of the properties of

TABLE OF PROPERTIES OF SATURATED STEAM.

Pressure in pounds per sq. in. above vacuum.	Temperature in degrees Fahrenheit.	Heat in liquid from 32° in units.	Heat of vaporization, or latent heat in heat units.	Total heat in heat units from water at 32°.	Density or weight of cubic ft. in pounds.	Volume of 1 pound in cubic feet.	Total pressure above vacuum.
1	101.99	70 0	1043.0	1113.1	0.00299	334.5	1
2	126.27	94.4	1026.1	1120.5	0 00576	173.6	2
3	141.62	109.8	1015.3	1125.1	0.00844	118.5	3
4	153.09	121.4	1007.2	1128.6	0.01107	90.31	4
5	162.34	130.7	1000.8	1131.5	0 01366	73 21	5
6	170.14	138.6	995.2	1133.8	0.01622	61 67	6
7	176.90	145.4	990.5	1135.9	0.01874	53.37	7
8	182.92	151.5	986.2	1137.7	0.02125	47.06	8
9	188.33	156.9	982.5	1139.4	0.02374	42 12	9
10	193.25	161.9	979.0	1140.9	0.02621	38.15	10
14 7	212.00	180.9	965.7	1146.6	0.03794	26.36	14.7
15	213.03	181.8	965.1	1146.9	0.03826	26.14	15
20	227.95	196.9	954.6	1151.5	0.05023	19.91	20
25	240 04	209.1	946.0	1155.1	0.06199	16.13	25
30	250.27	219.4	938.9	1158 3	0.07360	13.59	30
35	259.19	228.4	932.6	1161 0	0.08508	11.75	35
40	267.13	236.4	927.0	1163.4	0.09644	10.37	40
45	274.29	243.6	922.0	1165 6	0.1077	9 287	45
50	280.85	250.2	917.4	1167.6	0.1188	8 414	50
55	286.89	256.3	913.1	1169.4	0.1299	7.696	55
60	292.51	261.9	909.3	1171.2	0.1409	7.097	60
65	297.77	267.2	905.5	1172.7	0.1519	6 583	65
70	302.71	272.2	902.1	1174.3	0.1628	6.143	70
75	307.38	276.9	898.8	1175.7	0.1736	5.762	75
80	311.80	281.4	895.6	1177.0	0.1843	5 426	80
85	316.02	285.8	892.5	1178 3	0.1951	5.126	85
90	320.04	290.0	889.6	1179 6	0 2058	4.859	90
95	323.89	294.0	886.7	1180.7	0 2165	4.619	95
100	327.58	297.9	884.0	1181.9	0.2271	4.403	100
105	331.13	301.6	881.3	1182.9	0.2378	4.205	105
110	334.56	305.2	878.8	1184.0	0.2484	4.026	110
115	337.86	308.7	876.3	1185.0	0.2589	3.862	115
120	341.05	312.0	874.0	1186.0	0.2695	3 711	120
125	344.13	315.2	871.7	1186.9	0.2800	3.571	125
130	347.12	318.4	869.4	1187.8	0.2904	3.444	130
140	352.85	324.4	865.1	1189.5	0 3113	3 212	140
150	358.26	330.0	861.2	1191 2	0.3321	3 011	150
160	363.40	335.4	857.4	1192 8	0.3530	2 833	160
170	368.29	340.5	853.8	1194.3	0 3737	2.676	170
180	372.97	345.4	850.3	1195.7	0 3945	2.535	180
190	377.44	350.1	847.0	1197.1	0.4153	2.408	190
200	381.73	354.6	843.8	1198.4	0 4359	2.294	200
225	391.79	365.1	836.3	1201.4	0 4876	2 051	225
250	400 99	374.7	829.5	1204.2	0.5393	1.854	250
275	409.50	383.6	823.2	1206.8	0.5913	1.691	275
300	417.42	391.9	817.4	1209.3	0 644	1.553	300
325	424.82	399.6	811.9	1211.5	0 696	1.437	325
350	431.90	406.9	806.8	1213.7	0.748	1.337	350
375	438.40	414.2	801.5	1215.7	0.800	1.250	375
400	445.15	421.4	796.3	1217 7	0.853	1.172	400
500	466.57	444.3	779.9	1224 2	1.065	.939	500

TABLE OF PROPERTIES OF SATURATED STEAM.

Temperature in degrees Fahrenheit.	Total pressure above vacuum.	Heat in liquid from 32° in units.	Heat of vaporization, or latent heat in heat units.	Total heat in heat units from water at 32°.	Density or weight of cubic ft. in pounds.	Volume of one pound in cubic feet.	Temperature in degrees Fahrenheit.
32	0.089	0.	1091.7	1091.7	0.0003	3387.	32
60	0.254	28.12	1072.1	1100.2	0.0008	1234.	60
90	0.692	58.04	1051.4	1109.4	0.0021	474.6	90
120	1.683	88.10	1034.4	1118.5	0.0049	204.4	120
140	2.877	108.2	1016.4	1124.6	0.0081	123.2	140
150	3.706	118.3	1009.4	1127.7	0.0103	97.03	150
160	4.729	128.4	1002.3	1130.7	0.0130	77.14	160
170	5.98	138.5	995.3	1133.8	0.0162	61.85	170
180	7.50	148.5	988.3	1136.8	0.0200	50.01	180
190	9.33	158.6	981.3	1139.9	0.0245	40.73	190
200	11.52	168.7	974.2	1142.9	0.0299	33.40	200
210	14.12	178.8	967.2	1146.0	0.0363	27.57	210
220	17.19	188.9	960.1	1149.0	0.0435	22.98	220
225	18.91	193.9	956.7	1150.6	0.0476	20.99	225
230	20.78	198.9	953.2	1152.1	0.0521	19.20	230
235	22.80	204.0	949.6	1153.6	0.0569	17.59	235
240	24.98	209.0	946.1	1155.1	0.0619	16.14	240
245	27.33	214.1	942.6	1156.7	0.0674	14.83	245
250	29.86	219.1	939.1	1158.2	0.0733	13.65	250
255	32.57	224.1	935.6	1159.7	0.0795	12.57	255
260	35.48	229.2	932.0	1161.2	0.0862	11.60	260
265	38.60	234.2	928.6	1162.8	0.0933	10.72	265
270	41.94	239.3	925.0	1164.3	0.1008	9.918	270
275	45.51	244.3	921.5	1165.8	0.1088	9.187	275
280	49.33	249.3	918.0	1167.3	0.1173	8.521	280
285	53.39	254.4	914.5	1168.9	0.1264	7.913	285
290	57.72	259.4	911.0	1170.4	0.1359	7.356	290
295	62.33	264.4	907.4	1171.9	0.1461	6.847	295
300	67.22	269.5	903.9	1173.4	0.1567	6.380	300
305	72.42	274.5	900.5	1175.0	0.1680	5.952	305
310	77.83	279.6	896.9	1176.5	0.1799	5.558	310
315	83.77	284.8	893.2	1178.0	0.1925	5.195	315
320	89.95	290.0	889.5	1179.5	0.2058	4.861	320
325	96.48	295.2	885.9	1181.1	0.2197	4.552	325
330	103.38	300.5	882.1	1128.6	0.2343	4.267	330
335	110.66	305.7	878.4	1184.1	0.2498	4.004	335
340	118.34	310.9	874.7	1185.6	0.2660	3.760	340
345	126.43	316.1	871.1	1187.2	0.2830	3.534	345
350	134.95	321.4	867.3	1188.7	0.3008	3.324	350
355	143.91	326.6	863.6	1190.2	0.3195	3.130	355
360	153.33	331.8	859.9	1191.7	0.3391	2.949	360
365	163.22	337.1	856.2	1193.3	0.3597	2.780	365
370	173.60	342.3	852.5	1194.8	0.3812	2.623	370
375	184.49	347.5	818.8	1196.3	0.4038	2.476	375
380	195.91	352.8	845.0	1197.8	0.4276	2.338	380
385	207.87	358.0	841.4	1199.4	0.4521	2.212	385
390	220.39	363.2	837.7	1200.9	0.4780	2.092	390
395	233.50	368.4	834.0	1202.4	0.5051	1.980	395
400	247.21	373.7	830.2	1203.9	0.5336	1.874	400
405	261.55	378.9	826.6	1205.5	0.5633	1.775	405
410	276.54	384.1	822.9	1207.0	0.5945	1.682	410
415	292.21	389.4	819.1	1208.5	0.6270	1.595	415
420	308.57	394.6	815.4	1210.0	0.6610	1.512	420
425	325.65	399.8	811.8	1211.6	0.6970	1.434	425

saturated steam, one varying with the temperatures, the other with the pressures. The tables are made out in five unit intervals; intermediate points are proportional.

Example.— Suppose we wish to find the total heat corresponding to a pressure of 112.3 pounds (gage). We first add 14.7 to the 112.3 and get 127 pounds absolute pressure. The total heat of 1 pound of steam at 125 pounds pressure is 1,186.9. The total heat at 130 pounds pressure is 1,187.8.

Difference for 5 pounds = 1,187.8 — 1,186.9 = .9
Difference for 1 pound = .9 ÷ 5 = .18
Difference for 2 pounds = 2 × .18 = .36

The total heat for 127 pounds is:

Total heat at 125 pounds = 1,186.9
Difference for 2 pounds = .36
Total heat at 127 pounds = 1,187.26

This method is called interpolation, and in many complete tables the differences for the intervals are given to facilitate the work.

If steam tables are not at hand, there are several approximate formulas that may be used for rough calculations and estimates, but it must be borne in mind that results obtained by the use of these equations are not strictly accurate, and should not be used if the regular tables can be had.

Probably the relation of temperature and pressure will be most frequently needed. If the gage pressure is between 20 pounds and 100 pounds

$$t = 14\sqrt{p} + 198 \text{ approximately} \qquad [3]$$

where t = temperature in degrees Fahrenheit
p = gage pressure in pounds per square inch.

For pressures over 100 pounds per square inch (gage) we must modify the equation thus:

$$t = 14\sqrt{p} + 198 - \left(\frac{p - 100}{11}\right) \qquad [4]$$

These equations will cover a range of pressures from 20 to 340 pounds, and give an error of less than $1\frac{1}{2}$° in nearly all cases. From 35 pounds per square inch to 100 pounds the error is generally less than one-half of one degree.

For pressures below 20 pounds use the constant 196 instead of 198.

The latent heat may be approximately expressed by the formula,

$$l = 1{,}114 - .7\,t \qquad [5]$$

in which l = latent heat

t = temperature in degrees F.

This formula gives very good results for temperatures less than 320°, corresponding to a gage pressure of about 75 pounds. Above this pressure the formula gives slightly larger results than are found in the steam tables. At 250 pounds (gage) the formula gives 829.8 and the steam tables 825.8, so that the error will not be large in any case.

We have defined a B. T. U. as the amount of heat necessary to raise one pound of water from 61° F to 62° F. The specific heat of water is nearly constant over ordinary ranges of temperature and at 400° we find the heat of the liquid from the tables to be 373.7 B. T. U. By definition, the heat of the liquid at 32° is zero, so that a rise of 368° in temperature requires 373.7 B. T. U. If we consider the heat of the liquid proportional to the rise in temperature, our error will be 5.7 units of heat in 400°. At lower temperatures the error is much smaller, so that we may express the heat of the liquid approximately by the formula,

$$h = t - 32 \qquad [6]$$

in which h = the heat of the liquid

t = temperature in degrees as before.

The total heat is equal to the sum of the last two, or

$$\mathrm{H} = h + l \qquad [7]$$

The relation of pressure and volume of steam may be approximately expressed by the equation

$$\mathrm{P}\,\mathrm{V}^n = \mathrm{C}, \text{ or } \mathrm{V}^n = \frac{\mathrm{C}}{\mathrm{P}}$$

In which P = absolute pressure

V = specific volume

n = an exponent = $\frac{17}{16}$

C = a constant = 475 nearly.

We may then write the equation

$$V^{\frac{17}{16}} = \frac{475}{P} \text{ or } V = \sqrt[\frac{17}{16}]{\frac{475}{P}} \qquad [8]$$

This is called the equation of constant steam weight; it may be solved by the aid of logarithms.

The density can of course be determined from the specific volume.

Let us apply these approximate formulas to a specific case and see how the results compare with the actual quantities given in the steam tables. For this purpose we will suppose steam at 70.3 pounds gage pressure or 85 pounds absolute.

Equation (3) $t = 14\sqrt{p} + 198$

$= 14\sqrt{70.3} + 198 = 315.40°$

From steam tables, temperature $= 316.02°$

Equation (5) $r = 1{,}114 - .7\,t$

$r = 1{,}114 - (.7 \times 315.4) = 893.2$

From steam tables, latent heat $= 892.5$

Equation (6) $q = t - 32°$

$= 315.4 - 32 = 283.4$

From tables, heat in the liquid $= 285.8$

Equation (7) $H = q + r$

$= 285.8 + 893.2 = 1{,}179.0$

From tables, total heat $= 1{,}178.3$

Equation (8) $V = \sqrt[\frac{17}{16}]{\frac{475}{P}}$

$$V = \sqrt[\frac{17}{16}]{\frac{475}{85}} = 5.100$$

From tables, specific volume $= 5.125.$

A comparison of these results shows that these formulas ca not be used when accuracy is sought; but if only approxima results are desired they will be found satisfactory. Whenev possible the steam tables should be used in preference to a approximations.

Superheated Vapor. We have seen that a saturated vap contains just enough heat to keep it in the form of a vapor; if loses heat it will condense. A superheated vapor is one that has been heated after vaporization; it can lose this extra heat before any condensation will take place. A vapor in contact with its liquid is saturated; one heated after removal from the liquid is superheated.

For saturated steam there is a fixed temperature for every pressure. If we know either the pressure or the temperature, we can find the other in the steam tables. For instance, if the gage pressure of a boiler is 60.3 pounds and we wish to know the temperature, we simply add atmospheric pressure and turn to our tables and find it to be 307° (about).

With superheated steam the case is entirely different, for there is no longer the same direct relation between the temperature and pressure. In fact, the relation between temperature and pressure of superheated steam depends upon the amount of superheating. Superheated steam at 60.3 pounds gage pressure may have a temperature considerably above 307° F. At a given pressure the temperature and volume of a given weight of superheated steam are always greater than the temperature and volume of the same weight of saturated steam. The properties of superheated steam at given pressure are not constant as is the case with saturated steam.

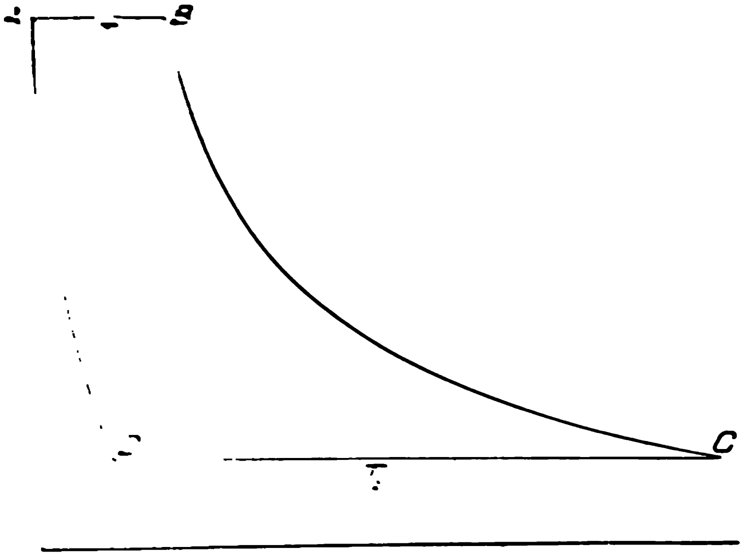
If superheated steam were a perfect gas, we could determine the relation of P, V and T by the equation $PV = CT$; but superheated steam is not a perfect gas, hence we must modify our equation. By experiment it has been determined that the following equation is nearly correct:

$$PV = 93.5\ T - 971\ P^{\frac{1}{4}}$$

In which P = absolute pressure in pounds per square foot
T = absolute temperature
V = volume of 1 pound in cubic feet.

THE STEAM ENGINE.

We have studied the action and formation of steam, and now we shall consider its application to the steam engine. We know that steam contains a great deal of heat, and that heat can be converted into work by allowing a working substance to pass from the high temperature of the heat generator to the lower temperature of the refrigerator, during this change giving up heat, which is transformed into work. There are several forms of heat engines, all of which convert the heat contained in some substance into work. At the present time the steam engine is the most important. When of good size and properly designed and run, it is as economical as any other heat engine, and it can be more easily controlled and regulated. We shall consider first the theoretically perfect engine and then the modifications that go to make up the steam engine of to-day.



The theoretical engine (Fig. 3) is supposed to receive heat from the generator at constant temperature T_1 until communication is interrupted at B. The working substance expands to C without losing or gaining any heat from external sources until the temperature of the refrigerator is reached. The engine now rejects heat at the constant temperature T_2 of the refrigerator and then compresses the working substance without loss or gain in the quantity of heat until the temperature of the heat generator is reached. These are ideal conditions, and, if fulfilled, the efficiency of the perfect engine will depend only on the difference between the temperature at which heat is received and rejected, or, in other words, it depends only upon the difference in temperature between the generator and the refrigerator.

If T_1 = absolute temperature of heat received and

T_2 = absolute temperature of heat rejected, then the thermal efficiency, E, of the engine will be represented by the formula,

$$E = \frac{T_1 - T_2}{T_1}$$

Or, in other words, the efficiency equals the absolute temperature of the heat rejected, subtracted from the absolute temperature of the heat received, and the remainder divided by the absolute temperature of the heat received.

Suppose an engine is supplied with steam at 120 pounds absolute pressure, and the exhaust is at atmospheric pressure. What is the thermal efficiency?

The absolute temperature corresponding to 120 pounds pressure is $341.05 + 461 = 802.05°$, and the absolute temperature of the exhaust is $212 + 461 = 673°$.

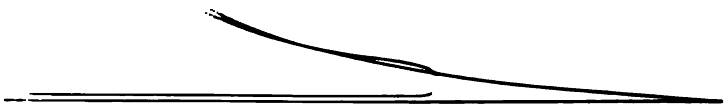
$$\text{Then} \quad E = \frac{802.05 - 673}{802.05} = .16, \text{ or 16 per cent.}$$

In actual engines this efficiency cannot be realized, because the difference between the heat received and the heat rejected is not all converted into useful work. Part of it is lost by radiation, conduction, condensation, leakage and imperfect action of the valves. The cylinder walls of the theoretical engine are supposed to be made of a nonconducting material, while in the actual engine the walls are of metal, which admits of a ready interchange of heat between cylinder and steam. This action of the walls cannot be overcome, and is so important that a failure to consider its influence will lead to serious errors in computations, and no design can be made intelligently if based on the theory of the engine with nonconducting walls. The theoretical engine carries on its expansion without the loss of any heat, while in the actual engine a large amount of heat is lost by radiation. There is also a considerable loss of pressure between the boiler and engine, due to resistance of flow through pipes and passages. In a slow-speed engine with large and direct ports and valves this trouble may be

[illegible]

[illegible]

[illegible]



[illegible]

[illegible] the end of the stroke, expan- [illegible] pressure at release is above [illegible] exhaust ports causes [illegible] the actual condenser pressure. [illegible] sweep the full length of the [illegible] must be filled with steam, which [illegible] The theoretical and actual cards are shown in Fig. 4.

EFFICIENCY OF THE ACTUAL ENGINE.

We have seen that the efficiency of the theoretical engine is purely a thermal consideration; the efficiency of the actual engine,

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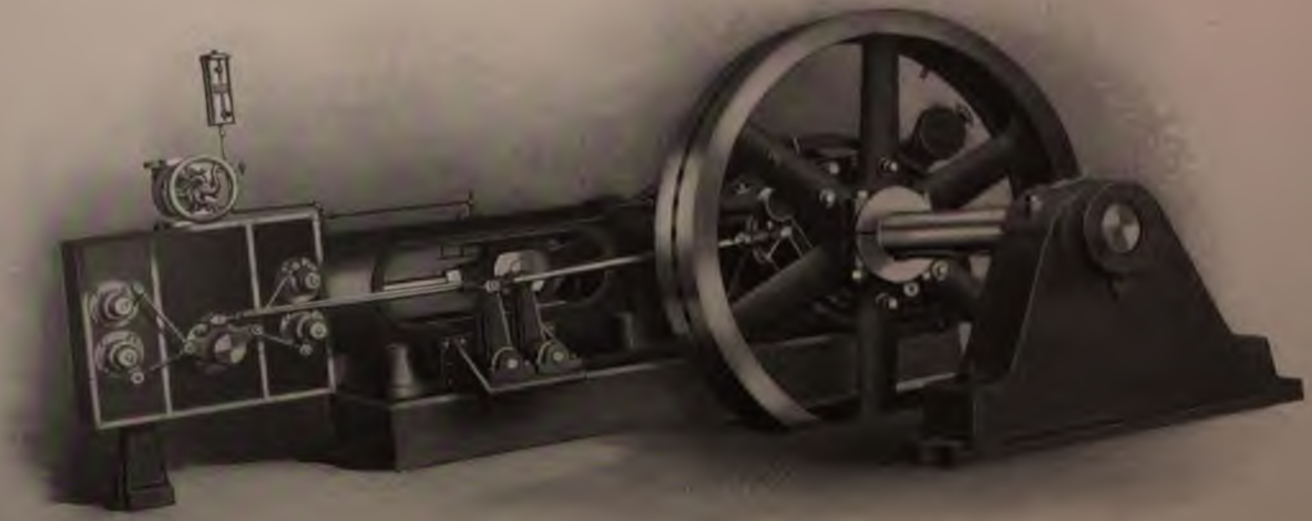

SIMPLE CORLISS VALVE ENGINE.
Ball and Wood Co.

owever, is a mechanical matter. The measure of the activity of
ork is the horse-power which corresponds to the development
f 33,000 foot-pounds per minute. As 778 foot-pounds are equiv-
ent to one B. T. U., 33,000 foot-pounds, or one horse-power, is
uivalent to 33,000 ÷ 778 = 42.42 B. T. U. Now if a certain
gine uses 84.84 B. T. U. per horse-power per minute, it is evi-
ent that its efficiency would only be $\frac{1}{2}$ or 50 per cent, because
2.42 ÷ 84.84 = $\frac{1}{2}$. Hence we may say that the efficiency of
e actual engine is equal to $\frac{42.42}{\text{B. T. U. per H. P. per minute}}$. This
fficiency is always much less than that of the perfect engine.
et us now discuss the effects of some of the losses.

In the first place, the metal, being a good conductor of heat,
ecomes heated by the steam within and transmits this heat by
nduction and radiation to the air or external bodies. With the
ylinder well lagged much less heat is lost by radiation. If the
gging were perfect and the temperature of the cylinder remained
e same as the temperature of the steam throughout the stroke,
ere would be no loss by radiation, but we should still lose heat
y conduction to the different parts of the engine.

During expansion, the temperature and pressure of the steam
ecrease as the volume increases, and the temperature at exhaust
much less than the temperature at admission. In the perfect
ngine the working substance after exhaust is compressed to the
emperature at admission, but in the actual engine much of this
team is lost and the compression of a part of it is incomplete, so
hat its temperature is less than the temperature at admission.

Suppose an engine is running with admission at 100 pounds
bsolute and exhaust at 18 pounds absolute. Then from steam
ables we find the temperature at admission to be 327.6°, and at
xhaust 222.4°. The metal walls of the cylinder, being good con-
uctors and radiators of heat, are cooled by the low temperature
f exhaust, so that the entering steam comes through ports and into
cylinder that is more than 100° cooler than the steam. This
eans that heat must flow from steam to metal until both are of
he same temperature. This causes the steam to give up part of
ts latent heat, and as saturated steam cannot lose any of its heat
ithout condensation, we find the cylinder walls covered with a

film of moisture known as initial condensation. This condensation in simple unjacketed engines working under fair conditions may easily be 25 per cent or more of the entering steam. The moisture in the cylinder has of course the same temperature as the steam; it has simply lost its heat of vaporization.

Although metal is a good conductor of heat it cannot give up nor absorb heat instantly; consequently during expansion the temperature of the steam falls more rapidly than that of the cylinder. This allows heat to flow from the cylinder walls to the moisture on them. As fast as the steam expands so that the pressure in the cylinder becomes less, this condensation will begin to evaporate. As the pressure falls it requires less and less heat to form steam, and therefore more and more of this moisture will be evaporated. At release the pressure drops suddenly, and more heat at once flows from the cylinder walls, and re-evaporation continues throughout the exhaust. Probably all of the water remaining in the cylinder at release is now re-evaporated, blows out into the air or the condenser, and is lost so far as useful work is concerned.

The steam that is first condensed in the cylinder does no work: its heat is used to warm up the cylinder, and later, when it is re-evaporated, it works only during a part of the expansion and at a reduced efficiency, because it is re-evaporated at a pressure and consequently at a temperature very much lower than that of admission. If the cut-off is short, perhaps 20 per cent of the steam condensed may be re-evaporated during expansion; if the cut-off is long, 10 per cent may be re-evaporated, the rest remaining in the cylinder at release still in the form of moisture. Thus some of the entering steam passes through the cylinder as moisture, until after cut-off, and still more passes clear through without doing any work at all.

Suppose an engine is using 30 pounds of steam per horse-power per hour and admission is at 100 pounds absolute. The latent heat of vaporization at this pressure is 884 B. T. U. per pound. If the condensation amounts to 33⅓ per cent, then 10 pounds are condensed and we lose 10 times 884, equals 8840 B. T. U. per hour, or 147.3 per minute; and since 42.42 B. T. U. represents 1 horse-power, we shall lose by condensation 147.3

divided by 42.42, equals $3\frac{1}{2}$ horse-power (nearly). If the cut-off is shortened, the condensation increases and may amount to 50 per cent at very short cut-off. Of course we use very much less steam at short cut-off than with long cut-off, and doubtless in many cases 50 per cent of the steam at short cut-off is not as great an absolute quantity as 30 per cent at long cut-off. Nevertheless, in all cases it is the percentages that go to make up the efficiency.

In addition to the actual loss from condensation in the cylinder there is still another loss due to the re-evaporation. Suppose, as before, that 10 pounds of steam are condensed in the cylinder, and that 20 per cent of this is re-evaporated during expansion. This will leave 8 pounds to be re-evaporated during exhaust. Suppose the exhaust is at 3 pounds above atmospheric pressure, or 18 pounds absolute (about). Then the heat of vaporization is 958.5 B. T. U. per pound of steam, and it will require 8 times 958.5, which equals 7668.0 B. T. U., to evaporate the 8 pounds. All of this heat is taken from the cylinder, leaving the engine much cooler than it would be were it not for this re-evaporation. This gives some idea of the great amount of heat passing away at exhaust, which is known as the exhaust waste.

In all cylinders it is necessary to have a little space between the cylinder cover and the piston when at the end of the stroke. In vertical engines the space is greater at the bottom than at the top. The volume of this space, together with the volume of the steam ports, is called the clearance. It varies from 1 to 20 per cent, depending upon the type and speed of the engine; the higher the speed, the greater the clearance. This clearance space must be filled with steam before the piston receives full pressure, but this steam does no work except in expanding, and the volume of the clearance offers additional surface for condensation.

Another important loss is that due to friction. We know that it takes considerable power to move an unloaded engine; if fitted with a plain, unbalanced slide valve, the power necessary to move the valve alone is considerable. The piston is made steam tight by packing rings, and leakage around the piston rod is prevented by stuffing boxes. All these devices cause friction as well as wear at the joints. The amount of power wasted in friction varies greatly, depending upon the kind of valves, general workmanship, state of repair and lubrication.

MULTIPLE EXPANSION.

Two engines may be used together on the same shaft, partly expanding the steam in one of the cylinders, and then passing it over to the other to finish the expansion. One advantage from this arrangement is that the parts can be made lighter. The high-pressure cylinder can be of much less diameter than would be possible if the entire expansion were to take place in one cylinder. This, of course, makes the pressure exerted on the piston rod much less, and the piston rod and connecting rod can thus be made much lighter. The low-pressure cylinder must be larger than it otherwise would be, but its parts need not be much heavier, because the pressure per square inch is always low.

This arrangement gives not only the advantage of lighter parts, but a decided increase of economy over the single cylinder type. If attention is given to the matter, a loss of economy would be expected, because the steam is exposed to a much larger surface through which to lose heat, but the gain comes from another source, and is sufficient to entirely counterbalance the effect of a larger cylinder surface.

When very high pressure steam and a great ratio of expansion is used, the difference between the temperature of the entering and that of the exhaust steam is great. For instance, suppose steam at 160 pounds (gage) pressure enters the cylinder and the exhaust pressure is 2 pounds (gage), the difference in temperature is, from steam tables,

$$370.5^\circ - 218.5^\circ = 152^\circ.$$

This difference becomes nearly 230° if the steam is condensed to about three pounds absolute pressure. The cylinder and ports of the engine are cooled to the low temperature of the exhaust steam, and, as we have seen, a considerable quantity of the entering steam is condensed to give up heat enough to raise the temperature of the cylinder to that of the entering steam. As the ratio of expansion increases, the difference in temperature increases, and consequently the amount of steam thus condensed also increases. To keep this initial condensation as small as possible we must limit the temperature, that is, we must not have as great a difference between admission and exhaust. To do this we must divide the expansion between two or more cylinders.

It will be remembered that the great trouble Watt found with Newcomen's engine was its great amount of condensation, and he stated as the law which all engines should try to approach, "that the cylinder should be kept as hot as the steam which enters it." This is to avoid condensation when steam first comes in. If, instead of expanding the steam in one cylinder, we expand it partly in one and then finish the expansion in another, we shall have passed it out of the first cylinder before its temperature falls a great deal, and consequently the cylinder walls will be hotter than they would be if we had expanded it entirely in one cylinder. This would then reduce the amount of steam condensed. The importance of this may not be evident at first, but it makes a great difference in the economy of the engine. If there is less condensation, there will be less moisture to re-evaporate, and consequently less exhaust waste; hence we shall save in two ways at the same time.

In a compound engine we admit the steam first to the smaller or high-pressure cylinder, and exhaust it to the larger or low-pressure.

Suppose steam at 160 pounds (gage) pressure is admitted to a cylinder, and the ratio of expansion is such that the steam is exhausted at about 60 pounds (gage) pressure; then the difference of temperature is

$$370.5° - 307° = 63.5°.$$

If, now, the steam when exhausted from the first cylinder enters a second and is allowed to complete its expansion, so that the exhaust pressure is about two pounds (gage) pressure, the difference of temperature in this cylinder will be

$$307° - 218.5° = 88.5°.$$

Then for the single engine, if the exhaust pressure is two pounds (gage), the difference of temperature is 152°, while in the compound engine this difference is divided into two parts,

.5° and 88.5°. The cylinder condensation for both cylinders the compound engine will be much less than if the total expan-n took place in a single cylinder. The cylinders should be so oportioned that the same quantity of work may be done in ch.

If there are two stages of expansion, the engine is called com-ound, three stages triple, and four quadruple.

Besides reducing the excessive condensation, there is still nother gain in using multiple expansion. We have seen how uch heat is lost by the exhaust waste, which in the simple ngine blows into the air or into the condenser, and is entirely lost. n the multiple expansion engine the exhaust and re-evaporation om one cylinder passes into the next and does work there; any akage from the high pressure is also saved, and does work in he low.

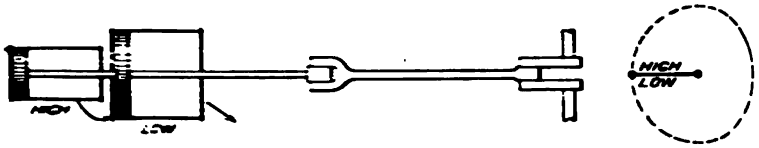

Fig. 5.

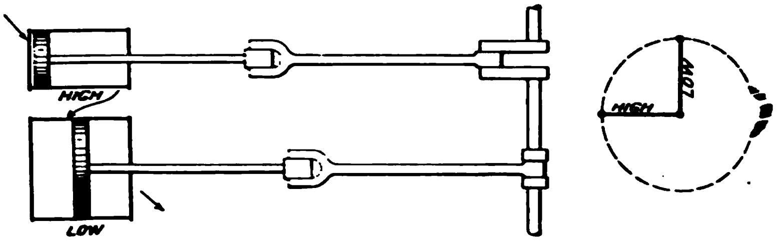

Fig. 6.

Mechanically there is a decided advantage; the several cranks ive a more even turning moment, and the distribution of work etween two or more cylinders makes it possible to use lighter ndividual parts; but there is a disadvantage in having more parts o look after, and a greater first cost of the engine.

Engines having two or more cylinders are arranged in various ways, but only the most common methods will be shown here. For two-cylinder compound engines the cylinders are often placed tandem, as shown in Fig. 5. This has the advantage of having but one crank, connecting rod and crosshead, but it has the disadvantage of dead points. In other words, the turning moment on the shaft is no more uniform than for a simple engine. When the cylinders are placed side by side and the cranks are at right angles, as shown in Fig. 6, the low-pressure cylinder cannot receive the exhaust steam directly from the high-pressure cylinder; consequently a receiver must be used. In this case the advantage is due to the even turning moment, and the disadvantage is from the cost of receiver, extra crosshead, crank and connecting rod. In many instances, as electric lighting, marine work, etc., the advantage of the more nearly constant force acting on the shaft is worth far more than the extra first cost. The same may be said

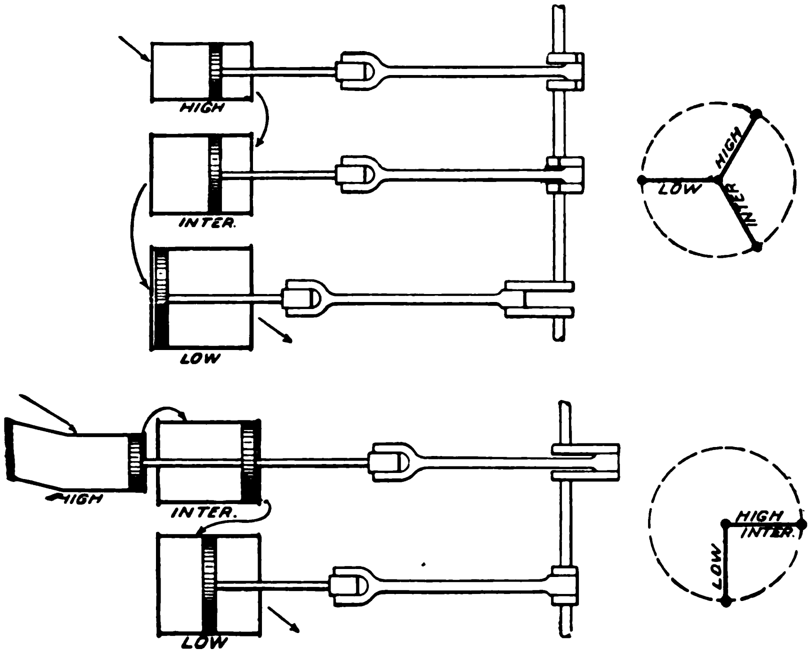

Fig. 7.

of triple and quadruple expansion engines. (See Fig. 7.) Each type of engine has its special service, and what is suitable in one case may not be serviceable in another. Fig. 8 shows the diagram of a quadruple expansion engine, which is more commonly used in marine work.

JACKETING.

Another method of reducing the loss due to cylinder condensation is to supply heat to the steam while it is in the cylinder.

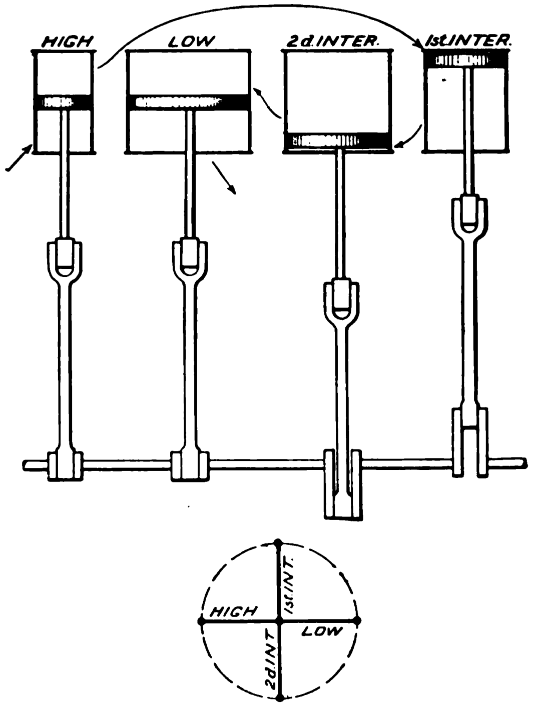

Fig. 8.

This is done by surrounding the cylinder with an iron casting and allowing live steam to circulate in the annular space thus formed. The cylinder covers are also made hollow to permit a circulation of live steam. A cylinder having the annular space (A, Fig. 9) filled with steam is said to be jacketed. A liner, L, is often used in jacketed cylinders.

The function of the jacket is to supply heat to the cylinder walls, to make up for that abstracted during expansion and exhaust, so that at admission the cylinder will be as hot as possible. The result is, that the difference in temperature between the cylinder walls and the entering steam is considerably less than in engines where no jacket is used. Condensation is therefore reduced, and since heat flows from the jacket to the cylinder during expansion, a much larger amount of this condensation is re-evaporated before release and it thus has a chance to do some work in the cylinder. This leaves a comparatively small amount of exhaust waste, and the heat thus abstracted is made good from the steam in the jacket. Since a large amount of heat is given up by the jacket steam a good deal of it must be condensed. Thus the question is asked: "What is the advantage of this method over that of allowing the entering steam to supply the heat by its own condensation?" This question is answered briefly as follows:

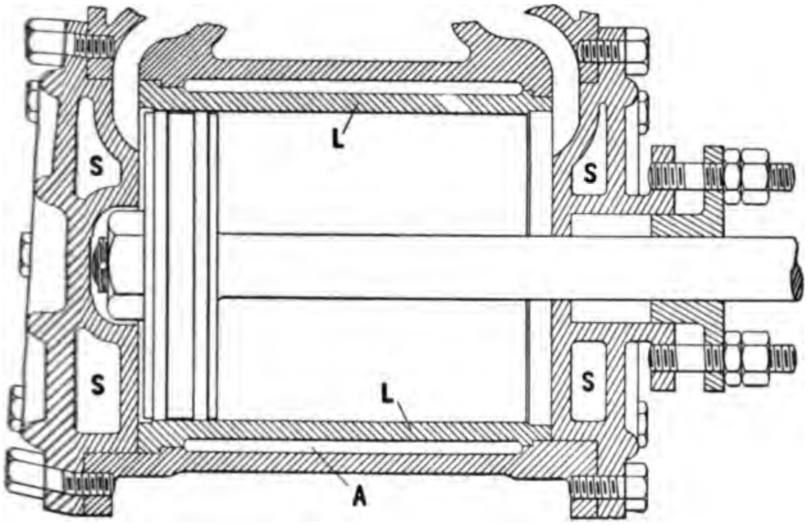

Fig. 9.

The loss of heat by condensing the steam would be less if the inside of the cylinder could be kept dry. We have seen how the moisture that collects by condensation is re-evaporated during expansion and exhaust, because the pressure falls and the cylinder walls are hotter than the steam. This re-evaporation takes place at the expense of the heat in the cylinder walls, and they are thus cooled. We have already seen that a great many B. T. U.'s are thus taken from the cylinder and thrown out at exhaust at every stroke. Now if we can keep the inside dry, so that there will be little or no re-evaporation at exhaust, we shall make a considerable saving. The steam that condenses in the jacket does not re-evaporate; it is returned to the boiler as feed water, so that the only heat lost is the latent heat given up during condensation. If the cylinder is heated from within, both the latent heat given up by condensation and the latent heat required for re-evaporation are lost.

In a triple expansion engine there is one distinct advantage in allowing condensation in the cylinder, for this moisture acts as a lubricant, and as the heat of re-evaporation passes into the next cylinder and there does work, there is very little loss. In many marine engines there are no means of lubricating the cylinders at all, the engineers depending entirely upon the condensation.

SUPERHEATED STEAM.

We have already learned that superheated steam contains more heat than is necessary to keep it in the form of steam, and that consequently it can part with some of that heat without condensing.

Suppose this superheated steam is admitted to the cylinder of an engine. We know that the cylinder walls would be comparatively cool, as they were in contact with the exhaust steam and were still further cooled by the re-evaporation of water which was condensed on them at the previous admission of steam. Heat must flow from the steam to the walls until they are as hot as the steam. If the steam were merely saturated, as is usually the case, some of it would be condensed in order to supply this heat to the walls. But in the case of the superheated steam we know heat was given to it beyond that necessary simply to make saturated steam, so that it can give up some heat to the walls before it begins to condense.

Thus the amount of initial condensation is materially reduced, and consequently there will be less cooling of the cylinder walls by the re-evaporation of this condensation; therefore the walls will be hotter at the next admission, and it will require less heat to raise them to the temperature of the incoming steam.

Besides the saving in the engine there is a considerable gain due to the increased efficiency of the boiler, for by superheating, a part of the heat in the waste gases is utilized. If a boiler primes or is so constructed that it does not furnish dry steam, the gain from superheating will be most marked. If there is moisture in the steam and this passes over into the engine, the heat that it contains is entirely wasted, and the re-evaporation of this additional moisture during exhaust causes the cylinder walls to cool off still further.

Suppose we have steam in a boiler at 60 pounds pressure by the gage. Its temperature will be about 307° F. If this steam leaves the boiler by a pipe which passes through a furnace or chimney where the temperature is greater than 307°, it will become heated as it passes through. The pressure will, however, be the same, for it is still in communication with the boiler. This is actually the way in which steam is superheated in practical work.

One great difficulty in superheating is that the superheater, which is usually placed in the uptake, is subjected to a very high temperature and there is great danger of its burning out. The plates of a boiler are not likely to be damaged by the intense heat of the furnace, because the water by its good conducting power prevents their becoming overheated. But steam is a poor conductor of heat, and although the superheater is placed in the uptake or chimney, where the heat is not as intense, they nevertheless cause a great deal of trouble.

Superheating is efficient if low pressure is used, that is, steam at about 50 pounds pressure; but if saturated steam is at 150 to 200 pounds pressure, the temperature is high. If more heat is then added, the temperature becomes such that lubricants are decomposed. In such cases the piston and valve, not being lubricated, cut into the cylinder walls and the valve seat, thus causing leaks and requiring excessive power to move the engine. The packing in the stuffing boxes, unless metallic, is injured and causes considerable trouble. In case high-pressure steam is used the superheating must be slight, often so slight as not to be worth the trouble and expense.

CONDENSERS.

In our study of the theoretical engine on page 20, we learned that to meet the ideal conditions and to attain the maximum efficiency, steam must enter the engine at the constant temperature of the heat generator (or boiler) and must leave the engine at the constant temperature of the refrigerator (or condenser). We have also learned that the difference between the temperature of admission and exhaust is a measure of the thermal efficiency. In Part I of "The Steam Engine," Watt's principle was stated, namely, that "when steam was condensed it should be cooled to as low a temperature as possible." In our discussion of saturated vapors we also learned that the temperature varied with the pressure, and consequently to cool the condensed steam to as low a temperature as possible simply means to condense the steam to as low a pressure as possible.

In the ordinary noncondensing engine, steam cannot be expanded below a pressure of 14.7 pounds, because the atmosphere exerts that pressure at the opening of the exhaust pipe. In fact, this 14.7 pounds is only the theoretical limit, and in practice the exhaust is always a little above this because of resistance in the exhaust ports and exhaust pipe; 17 or even 18 pounds back pressure is more nearly the conditions of actual service.

During the forward stroke, steam expands from the pressure at admission to a much lower pressure at release; then the valve opens for the return stroke and on one side of the piston there is full steam pressure, and on the other side the pressure of exhaust, which acts against the piston and against the force of the incoming steam. If all of this back pressure could be removed so that there would be a vacuum on the exhaust side of the piston, the power of the engine would be increased by just so many pounds of M. E. P., and in addition to this the steam could expand to a very much lower pressure and therefore work with greater economy.

One pound of steam at 17 pounds absolute pressure occupies 23.22 cubic feet of space in the cylinder of the engine, but one pound of water in the condenser occupies only about 0.016 cubic foot, which makes the steam occupy nearly 1,450 times as much space as the water into which it condenses. If, then, the exhaust steam could be condensed instantly, the back pressure would be reduced almost to zero and the engine would exhaust into a vacuum.

We know that a certain amount of heat is required to change one pound of water at a given temperature into steam at the same temperature; this is called the latent heat, or heat of vaporization. If the steam condenses, it must give up this latent heat. The easiest way of doing this is to let the steam mingle with a spray of water, as in the jet condenser, or come in contact with pipes through which cold water is circulated, as in the surface condenser. These two forms of condenser are fully described in Part I and will not be further considered here.

Unfortunately the mere condensation of the steam will not give a perfect vacuum, because more or less air, which is always in the water, comes over from the boiler and thus gets into the condenser. Moreover, the condensed water is hot, and a vapor rises from it in the condensing chamber; this, together with the air and some leakage, would spoil the vacuum were it not for the air pump, which removes the air and condensed steam. With the best air pump it would be impossible to maintain a perfect vacuum, but a vacuum of 26 inches, which corresponds to about 2 pounds absolute pressure, can readily be maintained in good practice.

Advantages of Condensing. It has already been stated that there is a gain in thermal efficiency by running an engine condensing, but it will be more clearly seen by considering a few figures. The thermal efficiency may be expressed by the formula:

$$E = \frac{T_1 - T_2}{T_1}.$$

This efficiency may be increased by making T_1 larger, which would happen if the boiler pressure were increased, or by making T_2 smaller, which would result from reducing the back pressure by condensing. If the boiler pressure is raised, both the numerator and denominator of the fraction will increase, and the value of the fraction will be but slightly greater. If, however, the back pressure is reduced, the numerator, $T_1 - T_2$, will be larger, while the denominator, T_1, will remain the same. It is apparent that this will cause a much greater increase in efficiency than raising the boiler pressure a like amount.

Suppose an engine is supplied with steam at 85.3 pounds (gage) pressure and it exhausts at 3.3 pounds (gage) pressure. The absolute temperature corresponding to $85.3 + 14.7 = 100$ pounds pressure is $327.58 + 461 = 788.58$, and the absolute temperature corresponding to $3.3 + 14.7 = 18$ pounds pressure is $222.40 + 461 = 683.40$. Then the thermal efficiency will be from the formula:

$$\frac{T_1 - T_2}{T_1} = \frac{788.58 - 683.40}{788.58} = .133 \text{ or } 13.3 \text{ per cent.}$$

If the boiler pressure were raised to 140 pounds absolute the efficiency would be

$$\frac{813.85 - 683.40}{813.85} = .16 \text{ or } 16 \text{ per cent.}$$

If instead of increasing the boiler pressure a condenser is used and thereby the exhaust pressure reduced to 4 pounds (absolute), the efficiency becomes

$$\frac{788.58 - 614.09}{788.58} = .221 = 22.1 \text{ per cent.}$$

Thus we see that if we lower the exhaust pressure 14 pounds we get a greater increase in efficiency than if the boiler pressure is raised 40 pounds.

Another method of showing the advantage by condensing the exhaust is by the indicator card.

Fig. 10 represents a card from a 12″ × 20″ engine making 75 revolutions with 75 pounds steam pressure. The dotted diagram represents a card taken when running without a condenser. Cut-off occurs at $\frac{1}{3}$ the stroke. The M. E. P. is 44.2 pounds. Hence the indicated horse-power is about 37.87.

The card shown in full line was taken with the same load, same speed, and with a condenser producing a vacuum of about 26 inches of mercury, which is equal to about 12.7 pounds. The absolute pressure at exhaust is then $14.7 - 12.7 = 2$ pounds. Since the load is the same, the areas and the M. E. P. must be the same in both cases. Cut-off is found to be only about $\frac{1}{6}$ the stroke. The above engine without the condenser uses an amount of steam represented by the length A C, while if a condenser is attached the steam used is represented by the length A C′. Thus we see that the amount of steam consumed per stroke is considerably less if a condenser is used.

Quantity of Water. Besides merely condensing the steam the injection water cools it still further, so that more than merely the latent heat is removed from it. If exhaust steam enters the condenser at a temperature t_1, it contains a certain amount of heat, which is the *total heat* at that temperature. If it is condensed and cooled to a temperature t_2, at which it leaves the condenser, it then contains a certain amount of heat which is the *heat of the liquid* at this temperature t_2.

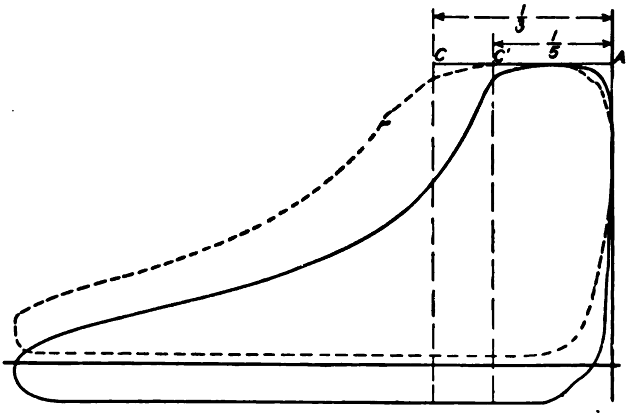

Fig. 10.

If A represents the total heat at t_1, and B represents the heat of the liquid at t_2, then the heat given up by one pound of condensed steam is equal to (A — B) B. T. U., provided the exhaust that enters the condenser is dry saturated steam. If C is the temperature of the injection water, and D is the temperature of the discharge water, then every pound of cooling water absorbs one B. T. U. for every degree rise in temperature; or we may say that the heat absorbed is equal to (D — C) B. T. U. per pound of cooling water. Then it will take as many pounds of water to absorb (A — B) heat units as (D — C) is contained times in (A — B). We may express this in terms of a formula thus:

$$W = \frac{A - B}{D - C}$$

In which W = pounds of cooling water per pound of steam.

For example: Suppose steam is expanded in an engine to 4 pounds absolute pressure. If the temperature of the injection water is 45°, and the condenser is of the surface type with discharge water at 120°, and the temperature of the condensed steam is 130°, how many pounds of injection water are required per pound of steam?

By consulting the steam tables we find the total heat of steam at 4 pounds pressure to be 1,128.6 B. T. U. The heat of the liquid in the condensed steam at 130° is 98.1 B. T. U. Then

$$W = \frac{1,128.6 - 98.1}{120 - 45} = 13.74 \text{ pounds.}$$

Suppose steam at 6 pounds absolute pressure exhausts into a jet condenser. The temperature of the injection water is 50° and the discharge is 120°. How many pounds of water are necessary to condense 8 pounds of steam?

In the jet condenser the temperature of the condensed steam and discharge water is the same. We find from the steam tables that the total heat of steam at 6 pounds absolute is 1,133.8 B. T. U., and the heat of the liquid in the condensed steam at 120° is 88.1 B. T. U. Then, as before,

$$W = \frac{1,133.8 - 88.1}{120 - 50} = 14.94.$$

For 8 pounds it will take $14.94 \times 8 = 119.52$ pounds.

The above calculation cannot be relied upon to any great extent, for we seldom know the true conditions in the condenser, and it would be of little value to us if we did know, as the exact conditions will change considerably. In practice it is customary to allow for about twice as much water as the above calculation would require. These figures give us a fair idea of the necessary sizes of pipes and passages leading to the condenser, and give a basis for estimating the dimensions of the air pump.

Cooling Surface. The amount of surface required to condense the steam in surface condensers depends upon the efficiency of the metal, the condition of the tubes, the difference in temperature between the two sides and their thickness. There have been no satisfactory tests to determine the amount of cooling surface

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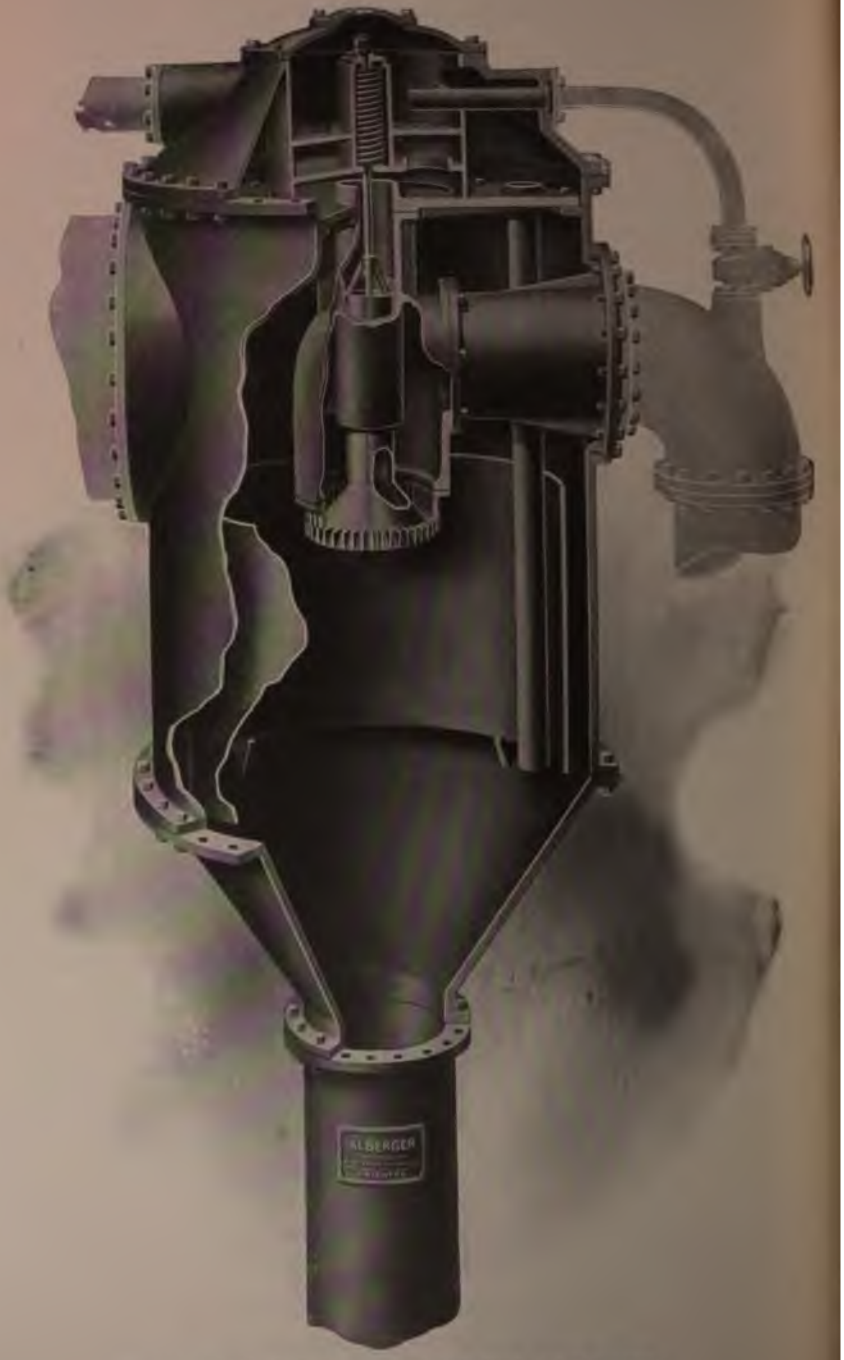


SECTIONAL VIEW OF BAROMETRIC CONDENSER.
Alberger Condenser Co.

necessary for a condenser, and in actual practice there seems to be a wide diversity of opinion. The tubes of a condenser are much thinner than boiler tubes, and much more clean, hence we might expect them to be more efficient in condensing the steam than the boiler tubes are in evaporating it. Boilers may evaporate 7 pounds or more of steam per hour, per square foot of heating surface. Seaton, an eminent authority on marine work, says that with cooling water at 60° F, and the discharge at 120°, a condensation of 13 pounds of steam per square foot per hour is fair average work. A new condenser will of course condense much more than this. If the exhaust pressure is from 6 pounds to 10 pounds absolute, an allowance of 1.5 to 1.8 square feet of cooling surface may be allowed per indicated horse-power, depending upon the pressure. This assumes that the temperatures of the injection and discharge water shall be 60° and 120° respectively.

It is evident that the amount of surface will depend upon the quantity of steam used per hour by the engine, the pressure and temperature of the exhaust and the temperature of the cooling water and discharge. There must also be an allowance for inefficient work after the condenser has become fouled with service. All these conditions make the problem so uncertain that calculations by means of formulæ are likely to be untrustworthy, and it is best at all times to make estimates from the figures given for similar conditions in actual service.

Measurement of Vacuum. We have seen that in order to maintain a vacuum in the condenser it is necessary to pump out, by means of an air pump, the air that leaks in. Evidently, if we are to maintain a proper vacuum, it is necessary to know at all times just how much pressure there is in the condenser. If the pressure increases, the air pump can be run a little faster until the proper vacuum is obtained.

From the study of pneumatics we know that the pressure of the atmosphere can be measured by means of a column of mercury. The atmospheric pressure will support a column of mercury about 30 inches high, which is equivalent to a pressure of 14.7 pounds (nearly). It would be inconvenient to attach a mercury column to the condenser and so we use a gage, in general appearance similar to a boiler gage except that the dial is graduated to read in inches of mercury instead of in pounds pressure, and it indicates the inches of vacuum ***below atmospheric pressure.*** If the pointer of the vacuum gage stands at 20, it means that the pressure is equal to 20 inches below atmospheric pressure. Since 30 inches is equal to 14.7 pounds, 20 inches would be equal to $\frac{20}{30} \times 14.7 = 9.8$ pounds. This is 9.8 pounds below atmosphere, or $14.7 - 9.8 = 4.9$ pounds above zero.

If the vacuum gage stands at 26 inches, what is the absolute pressure in the condenser?

$$\frac{26}{30} \times 14.7 = 12.74 \text{ pounds below atmosphere}$$

$$14.7 - 12.74 = 1.96 \text{ pounds absolute pressure.}$$

EXAMPLES FOR PRACTICE.

1. Steam enters the condenser at 11 pounds pressure (absolute). The water enters at 53° and leaves at 110°. The condensed steam leaves at 120°. If the engine uses 26 pounds of steam per horse-power per hour while running at 53 horse-power, what would be the theoretical amount of water used by the condenser and what should be the area of cooling surface of the condenser?

Ans. $\begin{cases} \text{106 square feet.} \\ \text{25,479 pounds.} \end{cases}$

2. Steam enters a jet condenser at 16 pounds absolute pressure. The injection water has a temperature of 48°. If the temperature of the discharge water is 115°, how many pounds of injection water are necessary per pound of steam?

Ans. 15.89 pounds.

3. If the pointer of a vacuum gage stands at 9, what is the approximate absolute pressure in the condenser?

Ans. 10.3 pounds.

4. The pointer of a vacuum gage which is attached to a condenser stands at 23. What is the approximate pressure in the condenser?

5. The steam gage indicates 175 pounds and the vacuum gage 26 inches of mercury. What is the total difference in pressure?

Ans. 187.74 pounds.

OTHER DEVICES.

Corliss Valves. From a thermal point of view the advantages of the Corliss valve may be summed up in a few words. The exhaust valves are separate from the admission valves and hence the exhaust steam does not come in contact with the admission ports. Thus the admission ports are not cooled and there is less condensation in proportion to the ratio of expansion than in the plain slide valve type of engine. The short ports reduce the volume of clearance and thus save clearance steam and reduce the surface exposed to condensation.

Separator. We have studied the loss of heat due to re-evaporation at exhaust and find that it is considerable. The more moisture there is in the entering steam, the more moisture there will be to re-evaporate; consequently more heat will be lost from the cylinder. It will at once be seen that if we can separate the moisture from the entering steam, we shall keep considerable water out of the cylinder.

PISTON SPEED.

The total distance passed over by the piston in one minute is called the piston speed. As the piston travels the length of the stroke twice for every revolution, the piston speed is evidently equal to the length of stroke multiplied by twice the number of revolutions. Then in the formula for horse-power, $\frac{PLAN}{33{,}000}$,

$$N = \text{the number of strokes}$$
$$L \times N = \text{the piston speed.}$$

In case the stroke is stated in inches, the piston speed in feet equals $\frac{L \times N}{12}$.

In determining the allowable piston speed, local conveniences, durability and the character of the work to be done should be considered. Other things being equal, the piston speed increases slightly with an increase in the length of stroke.

The piston speeds in common practice are as follows:

Direct acting pumping engines	100 to 150 feet per minute
Small stationary engines	300 to 500 feet per minute
Large stationary engines	500 to 900 feet per minute
Corliss engines	400 to 800 feet per minute
Locomotives	700 to 1,200 feet per minute
Marine engines	750 to 1,000 feet per minute

RATIO OF EXPANSION.

The ratio of expansion as usually understood is the piston displacement divided by the volume of the cylinder to the point of cut-off. Thus if the piston is 10 inches in diameter and the stroke is 12 inches, the piston displacement is $10^2 \times .7854 \times 12 = 942.48$ cubic inches. If cut-off occurs at $\frac{1}{3}$ the stroke, or when the piston has moved 4 inches, the volume to cut-off is $10^2 \times .7854 \times 4 = 314.16$ cubic inches, and the ratio of expansion is $\frac{942.48}{314.16} = 3$. This may be stated thus:

$$\frac{\text{area of cylinder} \times \text{length of stroke}}{\text{area of cylinder} \times \text{distance to cut-off}} = \text{ratio of expansion.}$$

Since the area of the cylinder appears in both numerator and denominator, we may cancel and write:

$$\text{Ratio of expansion} = \frac{\text{length of stroke}}{\text{distance to cut-off}}.$$

If clearance is taken into account, the true ratio of expansion is different from the apparent ratio as found above. The steam which expands in the cylinder is not merely the volume equal to the piston displacement, but is this volume plus the volume of the clearance.

With a late cut-off and small clearance the true ratio of expansion differs but little from the apparent. But when the cut-off is early and the clearance is large there is considerable difference, as shown in Fig. 11.

In this diagram D E represents the stroke, and is $2\frac{1}{2}$ inches long. D′ D represents the clearance volume, which is equal to $\frac{1}{8}$ the piston displacement, and is therefore .31 inch long. Let cut-off occur at $\frac{1}{8}$ of the stroke: then the apparent ratio of expansion is equal to $\frac{D\,E}{A\,C} = \frac{2\frac{1}{2}}{.31} = 8$. If we consider the clearance, the diagram is changed, as shown by the dotted lines, and the true ratio of expansion is

$$\frac{D'\,E}{A'\,C} = \frac{2.80}{.62} = 4.5.$$

Thus we see that the ratio of expansion is not 8 but 4.5.

WORK DONE IN THE CYLINDER.

The amount of work done per stroke in the engine cylinder is usually found by means of an indicator. It is often desirable, especially in designing engines, to know the work that may be expected under given conditions before the engine is built. The work done in the cylinder is proportional to the mean effective pressure, so that if we are to determine the probable power of our engine we must in some way ascertain the probable mean effective pressure. If the initial pressure, the back pressure and the ratio of expansion are known, we can find the probable M. E. P. by means of a formula. This formula does not provide for the losses due to compression, clearance, condensation and leakage, so that the value thus obtained must be multiplied by a factor depending upon the type of engine.

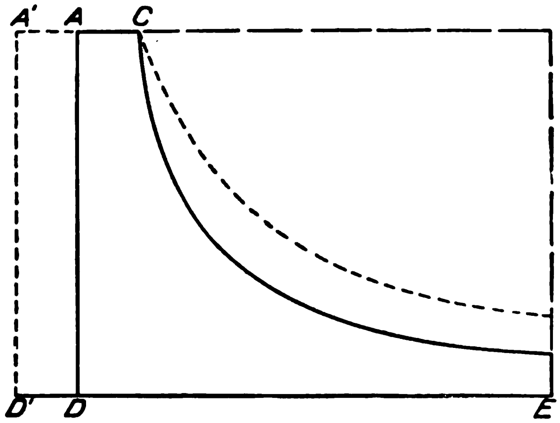


Fig. 11.

Let P = absolute initial pressure
R = ratio of expansion
p = back pressure (absolute)

Then neglecting all losses, the theoretical M. E. P. will be expressed by the formula,

$$\text{M. E. P.} = \text{P}\,\frac{(1 + \text{hyp. log. R})}{\text{R}} - p.$$

In order to allow for the losses due to compression, clearance condensation, wire-drawing, etc., it is customary to multiply value by factors given in the following table:

Engines with special valve gear and independent cut-off valves, above	.90
Plain slide-valve engines	.75 to .90
Compound engines	.65 to .80
Triple expansion engines	.50 to .70

These factors are for unjacketed engines; with jackets factor might be .05 more in each case. On account of wire-dr ing and large compression high-speed engines would have smaller value in its respective class. Only the best-desig engines with good valve gears and under favorable conditions warrant the use of the larger factor in their respective clas Large engines will of course have larger factors than small engi

Example.— What is the probable M. E. P. when the in pressure is 95.3 pounds (gage), the back pressure 3 pou (absolute) and the ratio of expansion 4? The engine is la of the plain slide-valve type with steam jackets.

$$P = 95.3 - 14.7 = 110 \text{ pounds}$$
$$p = 3 \text{ pounds}$$
$$R = 4$$

From the following table, hyp. log. R = 1.3863

$$\text{M. E. P.} = \frac{110\,(1 + 1.3863)}{4} - 3$$
$$= 62.62 \text{ pounds.}$$

For a large plain slide-valve engine, unjacketed, we migl expect a factor of .85 to .90, depending upon the conditions. Su pose we call the factor .87, then adding .05 for the effect of tl steam jacket we get .92.

62.62 × .92 = 57.6 pounds as the probable M. E. P.

A rough approximation to the power to be expected of compound, or triple expansion, engine, may be found in a simil manner by assuming that all the expansion takes place in the lo pressure cylinder, neglecting losses as before.

Example.— Find the probable I. H. P. for a triple expansic engine with the following dimensions and data:

Revolutions,	**115 per minute**
Initial pressure,	**140.3 pounds (gage)**
Back pressure,	**2 pounds (absolute)**
Total ratio of expansion,	**12**
Diameter of low-pressure cylinder,	**60 inches**
Stroke,	**30 inches**

$$\text{M. E. P.} = \frac{P\,(1 + \text{hyp. log. } R)}{R} - p$$

$$= 155\,\frac{(1 + 2.48490)}{12} - 2$$

$$= 43.01.$$

In practice only about .65 of this is actually obtained, so the probable M. E. P. is .65 × 43.01 = 27.95 pounds.

The I. H. P. is now found from the expression $\frac{P\,L\,A\,N}{33{,}000}$ as follows:

$$\text{I. H. P.} = \frac{27.95 \times 2\tfrac{1}{2} \times 2{,}827.4 \times 2 \times 115}{33{,}000} = 1{,}377.$$

TABLE OF HYPERBOLIC OR NAPIERIAN LOGARITHMS.

No.	Log.	No.	Log.	No.	Log.	No.	Log.
1.25	.22314	3.75	1.32175	6.25	1.83258	8.75	2.16905
1.50	.40546	4.00	1.38629	6.50	1.87180	9.00	2.19722
1.75	.55962	4.25	1.44692	6.75	1.90954	9.25	2.22462
2.00	.69315	4.50	1.50407	7.00	1.94591	9.50	2.25129
2.25	.81093	4.75	1.55814	7.25	1.98100	9.75	2.27726
2.50	.91629	5.00	1.60943	7.50	2.01490	10.00	2.30258
2.75	1.01160	5.25	1.65822	7.75	2.04769	12.00	2.48490
3.00	1.09861	5.50	1.70474	8.00	2.07944	15.00	2.70805
3.25	1.17865	5.75	1.74919	8.25	2.11021	18.00	2.89038
3.50	1.25276	6.00	1.79176	8.50	2.14006	20.00	2.99570

CRANK ACTION.

In the steam engine the steam exerts a pressure on the crank pin through the piston rod and connecting rod. When the crank is at the dead center, the entire pressure is on the bearing of the crank shaft; there is no tendency to turn the crank. In any other position the steam pressure tends to turn the crank pin. As the crank pin moves from dead center, the tendency increases until it reaches a maximum and then decreases until, at the other dead center, it is zero again. If the connecting rod were of infinite length, and steam were admitted throughout the whole stroke, the maximum tendency, or the maximum turning moment as it is called, would occur with the crank at right angles to the line connecting the dead points.

In the actual engine the thrust along the rod is constantly varying even though the pressure on the piston remains the same. This is due to the angularity of the connecting rod. The turning moment is always equal to the thrust along the connecting rod multiplied by the perpendicular distance from the connecting rod to the center of the shaft. If the steam pressure on the piston remains constant, the maximum turning moment occurs when the connecting rod is at right angles to the crank, for in this position the perpendicular distance from rod to center of shaft is a maximum equal to the length of the crank, and as the rod makes its greatest angle with the line connecting the dead center at this point the thrust along it will also be a maximum. If the cut-off is very early, $\frac{1}{4}$ stroke for instance, the maximum thrust along the rod will occur earlier than at the point previously mentioned, but the leverage of the force will be less, so that really there will be little change in the point of maximum turning moment no matter where the cut-off may occur.

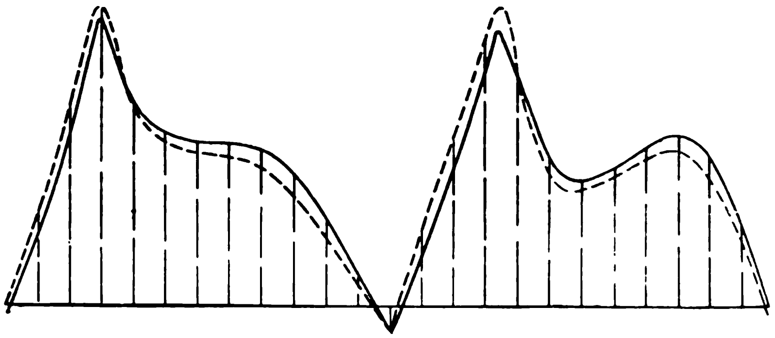


Fig. 12.

To represent this turning moment, diagrams of crank effort are drawn. These diagrams may be drawn with rectangular coordinates having the crank angles represented as abscissae and turning moments corresponding to these angles as ordinates. Besides the thrust of the connecting rod we must take into ount friction and the inertia of the reciprocating parts. At t this may be thought of small consequence, but with a fairly vy piston and connecting rod we can easily see that at high ed the momentum would be great. On a vertical engine, on up stroke the steam has to lift this heavy mass and impart a y considerable velocity to it, while on the down stroke the eleration of the mass is added to the steam pressure. This ces the effective force on the up stroke less than that due to actual steam pressure, and greater on the down stroke. The k effort diagram represented in Fig. 12 is from a horizontal ine of practical proportions. The initial steam pressure is 50 nds per square inch. Cut-off at $\frac{1}{6}$ stroke. The engine makes

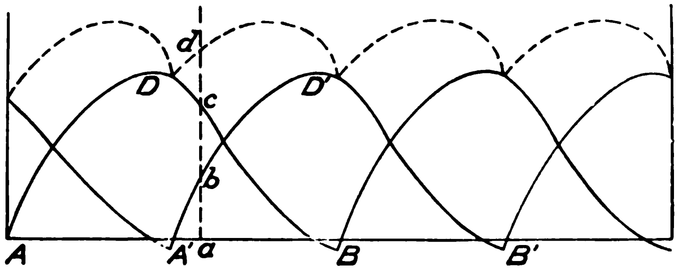


Fig. 13.

revolutions per minute. The dotted lines represent the k effort, without considering friction. The full line is the ram when weight of moving parts and inertia are considered. In drawing a crank effort diagram, friction is often neglected, the effect of inertia of the moving parts is of great importance, cially in the case of high speed.

It has been mentioned in connection with compounding that ere are two or more cranks on the shaft the turning moment ore nearly constant. We can now see that this is so. In 13, A D B represents the curve of turning moments on one k, and A′D′B′ the curve of turning moments on the other k, which is at right angles to the first. To find the total ing moment on the shaft the dotted line curve is drawn.

The points are obtained by adding the ordinates. Thus, to find the point b add [illegible]. Then $ab = a' - ai$. It is easily seen that the dotted line curve is more nearly a straight line than the curve A D B.

ECONOMY OF THE STEAM ENGINE.

The effects of various devices and methods employed to increase the economy of the steam engine cannot be studied from the theoretical side alone if we wish to obtain satisfactory results. So complicated and important is the action of the cylinder walls, that if we would learn the conditions favorable to economy we must study the actual tests of engines in service conditions.

Effect of Raising Steam Pressure. In general there is a gain in thermal efficiency by increasing the steam pressure and the total number of expansions, provided proper means are taken to lessen the thermal losses due to increased condensation and re-evaporation caused by the increased expansion. The initial condensation, however, places a pretty strict limit upon the number of expansions profitably used in a simple expansion engine.

If we raise the steam pressure and keep the same cut-off, the conditions will be slightly different. The losses from external radiation will increase with the rise of steam pressure, but the horse-power increases more rapidly. Hence there is a net gain, and since the changes of temperature are greater, the higher the pressure, it is reasonable to suppose that the loss from condensation and re-evaporation will increase. It is not difficult to imagine a point where these losses may offset the gain, and where, indeed, if the pressure is raised too far there will be a net thermal loss. This has actually happened in some cases. At first the gain from raising the steam pressure is rapid. As the pressure rises, the gain increases more slowly, and finally it is not worth the expense, if, indeed, there is not actual loss.

If we bear in mind that there is little gain in economy to be obtained by increasing the steam pressure beyond a certain moderate limit, unless the ratio of expansion is also increased, and that the losses from condensation, re-evaporation and exhaust waste limit the number of expansions profitably used in a simple expansion engine, we shall at once see that we must look to other devices for decided gains in economy. The most important of these have already been briefly considered.

Superheated Steam. Tests show a very decided gain in economy from using superheated steam, yet practically no permanent results have been obtained, on account of the difficulty of maintaining the superheating apparatus. To superheat to the best advantage we must have a coil of pipe in the flue and thus make use of the otherwise wasted heat. But the intense heat soon burns out the metal and together with the great pressure it is rapidly wasted away. Gains of from 15 per cent to 20 per cent from using superheated steam have actually been observed on simple engines. There has also been some gain noted by the use of superheated steam in triple expansion engines, but the gain is less than in the simple type, because naturally there is less condensation in the triple than in the simple engine.

Steam Jackets supply a small amount of heat to the cylinder during expansion which can be converted into work, but the chief advantage of a jacket is that it keeps the inner walls of the cylinder warmer at admission and thus reduces initial condensation and saves much loss of re-evaporation at exhaust. It will be evident that a large part of the heat of the steam jacket flows to the cylinder during exhaust and is thus entirely lost in the simple engine. In the triple engine this heat passes into the intermediate and low-pressure cylinders; consequently we might expect a greater gain from using a jacket on a triple than on a large simple engine. The main advantage of the jacket has been previously pointed out, and it may be stated that in all cases the gain from the jacket is small and there is to be found a considerable diversity of opinion as to its real advantages. On some engines there is undoubtedly little if any gain. The largest gain is in the smaller engines of say under 200 H. P.; on very small engines the gain is quite large, having been as much as 30 per cent on a 5″ × 10″ engine when developing only 1½ H. P. under light load. On a 10 H. P. engine the gain might be as much as 25 per cent. On engines of about 200 H. P. the gain would probably be 5 per cent to 10 per cent for simple condensing and compound condensing, and from 10 per cent to 15 per cent for triple expansion. The saving on large engines, of say 1,000 H. P., is very small, the reason being that large engines offer less cylinder surface per unit of volume than small ones, and hence we find proportionately less cylinder condensation in large engines than in small ones. The very small engines, in which the gain would be greatest, are seldom jacketed, because they are built for inexpensive machines and the first cost is of more consequence than the later economy.

Compounding is the most effective method of increasing the number of expansions and at the same time avoiding excessive cylinder condensation. We know that increase in boiler pressure and increase in expansion in simple engines are economical only to a certain limit. We shall now discuss the gain due to compounding and the conditions under which it is advantageous.

A direct comparison between tests of different engines is impossible because of the different steam pressures, etc., but a careful study seems to show that for simple condensing engines there is no advantage in raising the steam pressure above 80 pounds. In compounding, the pressure can be advantageously raised to 135 pounds. The gain due to the higher pressure, greater number of expansions and compounding may be 20 per cent to 30 per cent. For triple-expansion engines the most economical steam pressure is of course higher than for compound and may be used to advantage up to 180 pounds or more. The gain from using a triple over a compound may be about 5 to 10 per cent or more. These figures are good only for engines under full load and proper point of cut-off. A compound will usually suffer more loss of economy under light load than a simple, and the triple will suffer more than a compound.

Cut-off and Expansion. The best point for cut-off for a simple engine, whether jacketed or not, is about $\frac{1}{3}$ stroke if the engine is noncondensing and about $\frac{1}{6}$ stroke if condensing. The total expansions of a triple engine or a compound is commonly known as

$$\frac{\text{ratio of low pressure to high pressure}}{\text{fractional part of stroke at cut-off in high}}.$$

For instance, if the cylinders are as 1 : 3 : 8, and cut-off is $\frac{2}{5}$ stroke in high, then the total expansions as conventionally used is $\frac{8}{\frac{2}{5}} = 20$. For the best service of triple engines on land 20 expansions are used. The conditions of service make it impossible to use as many expansions in marine work, hence the relatively poor economy of marine engines. For compound engines 15 expansions seems to be the best in general. Of course it must be understood that the type of engine and conditions of service may necessitate different arrangements, but when such is the case a less economy of steam consumption usually results.

Variation of Load. An engine should be designed to give a fair economy over a reasonable variation of load. Ordinarily if it gives its best economy at normal load there will be a sufficient range. Usually, it happens that the best mechanical efficiency is obtained with a little longer cut-off than that which shows the most economical steam consumption per indicated horse-power. This is because, as the cut-off lengthens, the power increases faster than the frictional losses. It will be evident from this that increasing the cut-off slightly reduces the thermal efficiency, but the gain in mechanical efficiency may offset the thermal loss. Shortening the cut-off causes loss of both thermal and mechanical efficiency. It would then seem that if the engine were given a little longer cut-off than that which would produce the best thermal efficiency at normal load, the power could be reduced with less actual loss. For a slight reduction we should lose mechanical efficiency and gain thermal, and for a slight increase we should gain mechanical and lose thermal. Thus there would be a wider range with good results. There is always more loss by decreasing cut-off below the point of maximum efficiency than by lengthening it. That is, the engine works at a greater disadvantage when running under a light load than when running under a heavy one.

The allowable range of load for a simple engine is greater than for a compound or triple. If the power of a compound is reduced by shortening the cut-off of the high-pressure cylinder without shortening the low proportionally, there is likely to be an uneven distribution of work and, consequently, a wide fluctuation of temperature, which will cause so much condensation as to offset the advantages of compounding. Moreover, the large expansion in the high-pressure cylinder may reduce the admission pressure to the low cylinder so much that the expansion in the low may be carried below the atmosphere in a noncondensing engine and thus cause a loop at the end of the card, as we have learned in "Steam Engine Indicators." This loop means a loss of power, and that the high-pressure piston is dragging the low and the engine is using up power on itself as it were. A triple engine is even more troublesome than a compound, and besides giving the trouble already mentioned, the dragging of the low-pressure piston may injure the cylinder and loosen the guides. There is little difficulty in increasing the power of a compound or triple. A simple engine may easily be run at reduced power, either by shortening the cut-off or reducing the steam pressure or both.

Effect of Speed. The transfer of heat between the steam and walls of the cylinder, although very rapid, is not instantaneous. The longer the steam can remain in contact with the cylinder walls, the more heat will be lost. Hence it is reasonable to suppose that an increase of speed will reduce condensation. This has been found by tests actually to be the case. Other things being equal, a reasonable increase of speed results in better economy, but as we can seldom get as good a valve motion at high speed, we may lose almost as much in this way as we gain by decreasing our condensation. The most efficient valve gears are to be found on slow-speed engines, and hence we usually find that the most economical engines are slow speed. Nevertheless, with a given valve gear, an increase of speed gives better economy up to the limit of good valve motion. High-speed engines also require more clearance, which means another loss. These two factors, valve motion and clearance, limit our increase of speed beyond a certain point, just as condensation will limit our indefinite increase of boiler pressure.

Feed Water Heaters. The use of feed water heaters has been discussed in the instruction papers on boilers, but it may be well to observe here that in many places where water is expensive condensing engines cannot be run; nevertheless, a very considerable saving can be effected by allowing the exhaust steam to condense in a feed water heater, and thus save the heat that would otherwise be wasted, or the exhaust steam may be used for heating purposes. Of course in such cases the steam consumption of the engine is high, but if proper allowance is made for the heat used for other purposes, the actual fuel consumption rightfully charged to the engine is not excessive. If the feed water is heated by waste gases, then the gain belongs to the boiler and not to the engine.

The following represents the average steam consumption in pounds per I. H. P. per hour for various types of engine:

Triple expansion stationary	11 to 14 pounds
Triple expansion marine	13 to 16 pounds
Compound stationary	12 to 15 pounds
Compound marine	18 to 21 pounds
Simple condensing	17 to 20 with jacket
Simple condensing	18½ to 22 without jacket
Simple noncondensing	24 to 33 or more

All these figures have been taken from the results of careful tests of engines in actual service, and represent good engines of their respective types.

Direct-acting steam pumps, although very important engines, are extravagant in their consumption of steam. The absence of a fly wheel makes it almost impossible to use steam expansively, and hence the pump takes steam full stroke.

The steam consumption for a simple direct-acting steam pump may be anywhere from 60 to 200 pounds per horse-power per hour, depending upon the size and conditions of service.

The economy of an engine is usually stated in terms of steam per horse-power per hour, or of B. T. U. per horse-power per minute. The latter is more useful for purposes of comparison with other engine tests, but the real criterion of economy is the actual cost of power in terms of fuel consumption, for this can at once be reduced to dollars and cents, and the manufacturer or the builder of the engine can determine the effect of devices for economy in terms of money, which to him is worth more than B. T. U. It may often happen that some expensive device, while effecting a saving of heat consumption, may not save fuel enough to pay for its first cost, owing to the peculiar conditions of service.

TESTING.

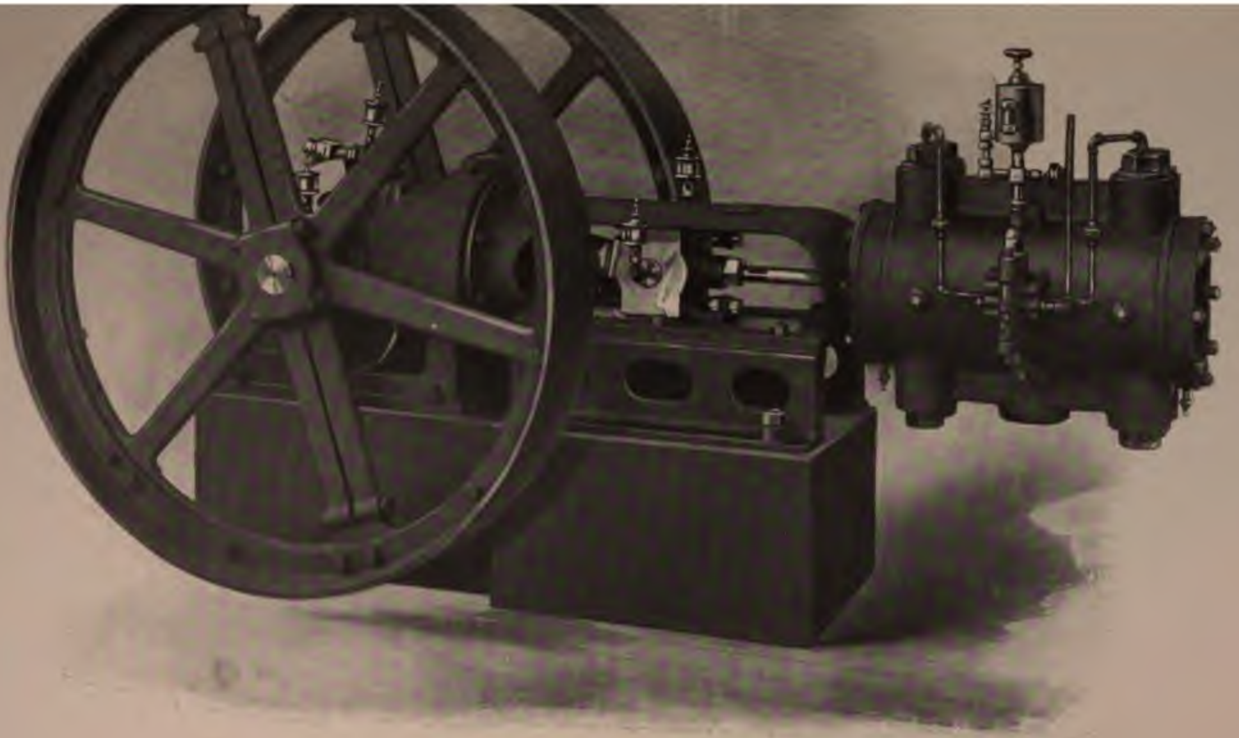
The principal object in testing a steam engine is to determine the cost of power or the effect of such conditions as superheating, jacketing, etc., upon the economy of the engine. We must therefore, in the first case, measure the cost of fuel, and in the second the actual heat used. In either case we must calculate the power of the engine.

The indicated power is determined in the familiar way by means of the indicator, and the actual power by means of a dynamometer or friction brake. (For further particulars of these instruments and apparatus see "Steam Engine Indicators.") To determine the cost of power in terms of coal it is necessary to conduct a careful boiler test, usually of twenty-four hours duration.

When the cost is expressed in terms of steam per horsepower per hour, we may follow either of two methods. We may condense and weigh the exhaust steam, or we may weigh the feed water supplied to the boiler. An hour under favorable conditions is usually sufficient for such tests. Steam used for any purpose other than running the engine must be determined separately and allowed for.

Probably the most accurate terms in which to state the performance of an engine is in B. T. U. per horse-power per minute. When the cost is expressed thus, it is necessary to measure the steam pressure, amount of moisture in the steam, and temperature of condensed steam when it leaves the condenser. Jacket stea must be accounted for separately. Important engines with thei boilers, etc., are usually built under contract to give a certai efficiency, and their fulfilment of this contract can be determine only by a complete test of the entire plant. Before beginnin the test, the engine should be run for a short time in order t limber it up and get it thoroughly warmed. It is of the utmos importance that all conditions of the test should remain constan especially the boiler pressure and the load. All instruments use in the test should be tested themselves before being used, in orde to determine the effect of any errors to which they may be subjec

Thermometers. All important temperatures, such as fee water, injection water, condensed steam, etc., must be taken b reliable thermometers, the errors of which have been previousl determined and allowed for. Good thermometers sold by reliab dealers are usually satisfactory. Thermometers with detachab scales are subject to serious errors, and should be used only f very crude work. Cheap thermometers are of little value in engine test.



BELT-DRIVEN AIR COMPRESSOR.
Sectionalized for Mule-back Transportation.
Sullivan Machinery Company.

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Indicators. The most important and least satisfactory instrument used in the test is the indicator. It is subject to an error of 2 per cent or 3 per cent at low speed, and this may easily be two or three times as much at high speed. It does not work satisfactorily at more than 400 revolutions per minute. If the indicator is carefully tested under conditions similar to those on the engine, the errors may be reduced to a minimum, but there will always be some uncertainty. The principal errors to which the indicator is subject have been mentioned in the instruction paper on "Steam Engine Indicators." It may, however, be well to add that for accurate work we should always use two indicators, as the long piping and joints necessary for only one causes a considerable loss of pressure and much condensation. For marine work it is customary to use only one indicator, with a three-way cock, the lower end of the cylinder usually being inaccessible.

Scales. Weighing should be done on standard platform scales. The water may be weighed in barrels provided with large drain valves which will allow the water to run out quickly. It is seldom possible to drain barrels completely, and so it is best to let out what will run freely, then shut the valve and weigh the barrel. This we call "empty" weight, and deducted from the weight "full," evidently gives us the true weight of water.

If not convenient to weigh the water, it may be measured in tanks or receptacles of known capacity, and the temperature taken, allowing the proper weight per cubic foot for water at that temperature; or it may be determined by meters.

Meters. Water meters are of two kinds, those that record the amount of water by displacement of a piston, and those in which the flow is recorded by means of a rotating disc. Piston water meters can be made very accurate, and if working under fair conditions of service they may be relied upon to a close degree. The chief error in a meter arises from the air that may be in the water. To reduce this error to a minimum, the meter should be vented, to allow the air to escape without passing through the meter. Rotary meters are good enough for very rough work, but are seldom sufficiently accurate for a careful engine test. So far as possible, weirs should not be used in engine work. They may be fairly accurate under certain conditions,

LOG OF TEST.

Time of Observation	Revolutions.	M. E. P.	PRESSURES.				TEMPERATURES.			WEIGHTS.		
			Boiler.	1st Receiver.	2d Receiver.	Condenser.	Feed Water.	Injection Water.	Condenser Discharge.	Coal.	Injection Water.	Discharge Water.

but a very little oil in the water may affect them seriously. They may sometimes be used to measure the discharge from a jet condenser, for then the volume is so large that the actual error is proportionately small.

Gages. Pressures should be measured on good gages that have been recently tested by comparison with a mercury column. The atmospheric pressure should be read from the barometer, and for accurate work this pressure should be used. For ordinary work, 30 inches, or 14.7 pounds, will do.

Calorimeter. When using superheated steam it is sufficient to take the temperature and pressure in the steam pipe, but if saturated steam is used, we must determine the amount of moisture it contains. This is done by means of a calorimeter such as described in "Boiler Accessories."

Brake. Any of the forms of friction brake described in 'Steam Engine Indicators" will answer the purpose. For smooth and continuous running it is essential that the brake and its band be cooled by a continuous stream of water. The water may either irculate in the rim of the wheel or around the brake band, but it must not come in contact with the rubbing surfaces.

If the load is steady, seven or eight observations at equal inervals will usually be sufficient. If possible, the cards should be aken simultaneously, and then all the data averaged for the final esult. If the load fluctuates, the cards must be taken oftener, nd a greater number of observations will be required. The greatest care and accuracy must be used in all this work. In conductng a test, a careful log should be kept of the data; the outline given on page 56 being a suggestion.

THE STEAM TURBINE.

A description of the steam turbine and the general principles of the engine were given in Part I of "The Steam Engine." Now we shall discuss its efficiency. The turbine is such a comparatively new engine that there have been but few tests made, and consequently we know very much less about its possibilities than is the case with the ordinary reciprocating engine.

Probably the greatest loss in the reciprocating engine is that due to condensation and the subsequent re-evaporation at exhaust.

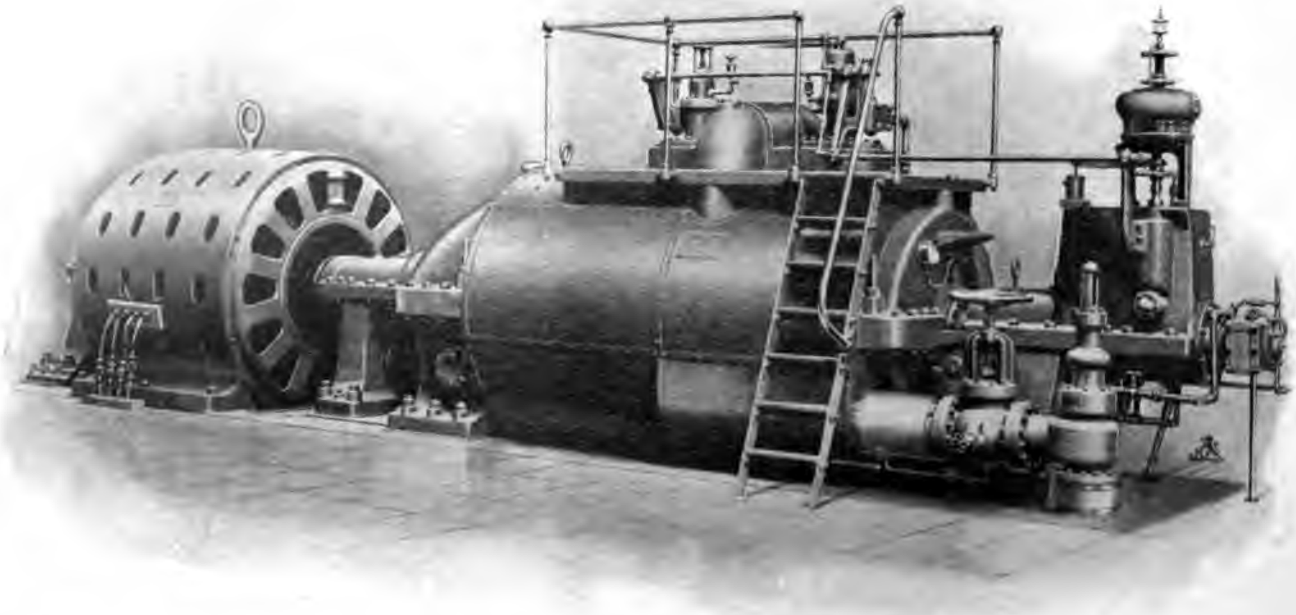
The exhaust cools the cylinder, so that the incoming steam n cool walls, while in the turbine no such conditions exist. admission takes place at one end of the engine and the exh occurs at the other end. The temperature gradually falls : admission to exhaust and the expanded steam never come contact with the part in which the higher-pressure steam w Thus Watt's principle seems to be fulfilled, namely, "The cyli should always be as hot as the steam that enters it." Of co there is considerable loss from radiation, but there should be much less condensation than in the reciprocating type.

Superheating may be made use of with considerable ga economy. There are no rubbing surfaces, no lubricants to de pose and no glands to burn out, as is the case in the reciproc engine. The steam may be superheated to 60° or more to ad tage. There being no internal lubrication, there is of course n to get into the condenser, and so the feed water may be used out fear of getting grease into the boiler.

Another advantage seems to be in the more complete ex sion of the steam. There is little gain in the reciprocating en by expanding the steam beyond a certain limit, because of th creased condensation. The boiler pressure cannot be incr indefinitely, neither can the expansion be carried out to the

From these considerations it would seem as if the tu ought to show much better efficiency than the reciprocating en and were it not for the friction of steam against the vanes o turbine the advantage would doubtless be in its favor. Tests shown a consumption of about 16 pounds of steam per B. per hour. Assuming an efficiency of 85 per cent, this would about 14 pounds per I. H. P. per hour. Tests of the best mo triple expansion pumping engines have shown a steam cons tion of a little over 11 pounds per I. H. P., and numerous tes ordinary triple expansion engines have been made which sh consumption of 12 to 13 pounds.

The most recent test of which we have accurate knowl was made on a Westinghouse-Parsons engine of 600 H. P. running at 3,600 revolutions per minute. With ordinary s having 3 per cent of priming, the turbine used 15.5 poun steam per B. H. P. per hour with a vacuum of $25\frac{7}{8}$ inches in condenser. With 30° of superheating and $26\frac{1}{2}$ inches of vacuum, the steam consumption fell to 14.2 pounds. Assuming 85 per cent mechanical efficiency as before would give a relative steam consumption of 13.17 pounds per I. H.P. in the first case, and 12.07 pounds in the second case. It will be instructive to compare these figures with those given for reciprocating engines on page 53.



WESTINGHOUSE-PARSONS TURBINE-GENERATOR UNIT OF 1,000-KILOWATT RATED CAPACITY AS IT APPEARS WHEN INSTALLED.

THE STEAM TURBINE.

The principle of the Curtis turbine differs from that of any other type in that it permits the use of moderate rotative speeds and very compact and simple mechanism. The turbine is divided into stages, each of which contains one, two, or more, revolving

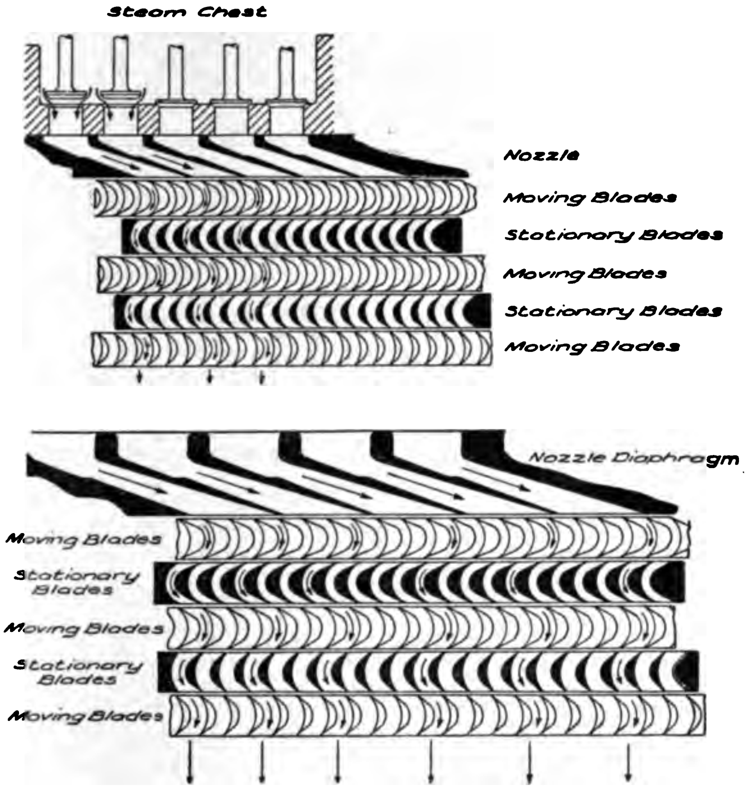
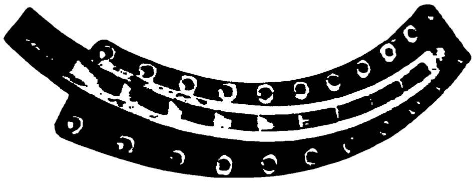


DIAGRAM OF NOZZLES AND BUCKETS IN CURTIS STEAM TURBINE.

uckets supplied with steam from a set of expansion nozzles. As he work is divided into several stages, the nozzle velocity in each tage is reduced, thereby rendering the nozzle action more efficient nd perfect than is possible where a higher initial velocity is mparted. The division of pressure between the stages is arranged o utilize the largest possible proportion of the energy of expanion. The position of the moving and stationary buckets with elation to the nozzle is shown in the accompanying diagram.

Vertical Type. For turbines of large capacity the General Electric Company has applied these principles to a turbine with a vertical shaft to avoid all imposition of weight on cylindrical bearings and tendency to shaft deflection as well as all difficulties due to irregularity of expansion or imperfections of support. The turbine is compact and of the greatest mechanical simplicity.

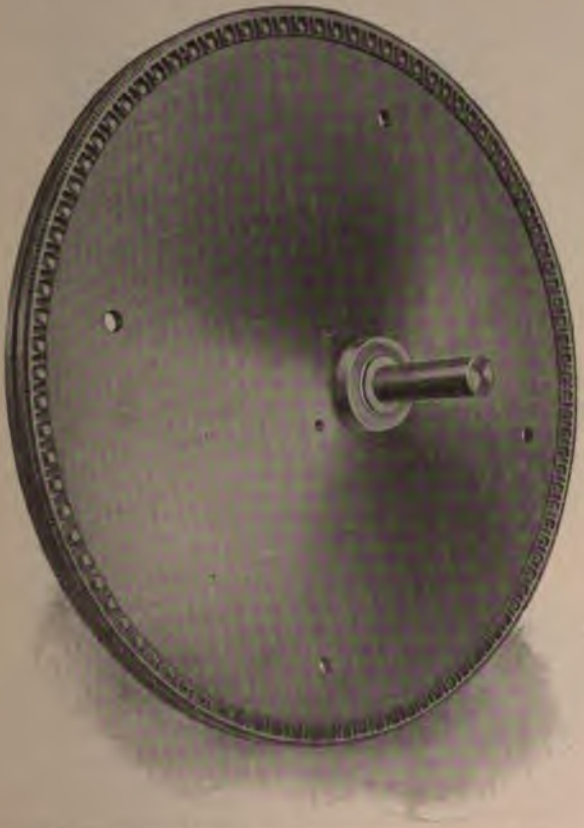
Step-Bearing. The step-bearing at the end of the vertical shaft supports the revolving part and maintains the revolving and stationary elements in exact relation. It consists of two cylindrical cast-iron plates bearing upon each other and with a central recess to receive the lubricating fluid, which is forced in by pumps with a pressure sufficient to sustain the weight of the revolving part. It is apparent that the entire weight of the machine is thus carried upon a film of lubricating fluid and that there is no appreciable friction. When the flow of liquid is interrupted, the bearing is slowly worn away, but experience has shown that interruptions in the flow seldom cause any deterioration which prevents the continuance of the machine in service after the flow is re-established. The tendency of the bearing in such cases is to wear itself to a new surface so that it operates normally.



NOZZLE

All large steam turbines are necessarily dependent upon forced lubrication. Failure of lubrication in a horizontal turbine is liable to cause serious trouble through cutting of the shaft or interference with the alignment. In the Curtis vertical type the possibility of any trouble is minimized and the simple cast-iron blocks can readily be replaced at trifling expense. In large stations, where several turbines are operated, it is desirable to install weighted accumulators which will maintain a constant pressure and also act as a reserve.

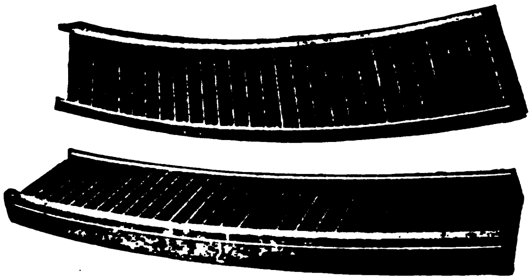
Clearances. In consequence of the exact relation maintained between the revolving and stationary elements by the step-bearing, it is possible to operate the turbines with very small clearances between the moving and stationary buckets. Experience, however, has shown that the reduction of clearance beyond a certain point, is not beneficial, and that clearances less than those which are desirable for economical reasons can be used without mechanical difficulty.



REVOLVING WHEEL FOR 2000 KW. CURTIS STEAM TURBINE.

Balance. A most important matter in all steam turbine work is the balance; and the importance of good balance applies as well to vertical turbines as those that are operated in a horizontal position. When the balance is good, the bearings on the vertical turbine shaft are practically free from strain or friction. It is possible to operate these machines successfully with a considerable imperfection of balance, but a perfect balance is practicable and should be attained in every case.

Governing. The speed of these turbines with variable load is controlled by the automatic opening and closing of the original admission nozzle sections. A centrifugal governor, attached to the top of the shaft, imparts motion to levers which in turn work the valve mechanism. There are several valves, each communicating with a single nozzle section, or in some cases two or more nozzle sections. These valves are connected to long pistons, by which the valve can be opened or closed by steam. The motion of each of these pistons is controlled by a small pilot valve which is worked by the governor mechanism. The movement of the governor mechanism moves the pilot valves successively and the main valves are opened or closed by the steam. By suitable adjustment, almost any degree of accuracy in speed control is obtainable.

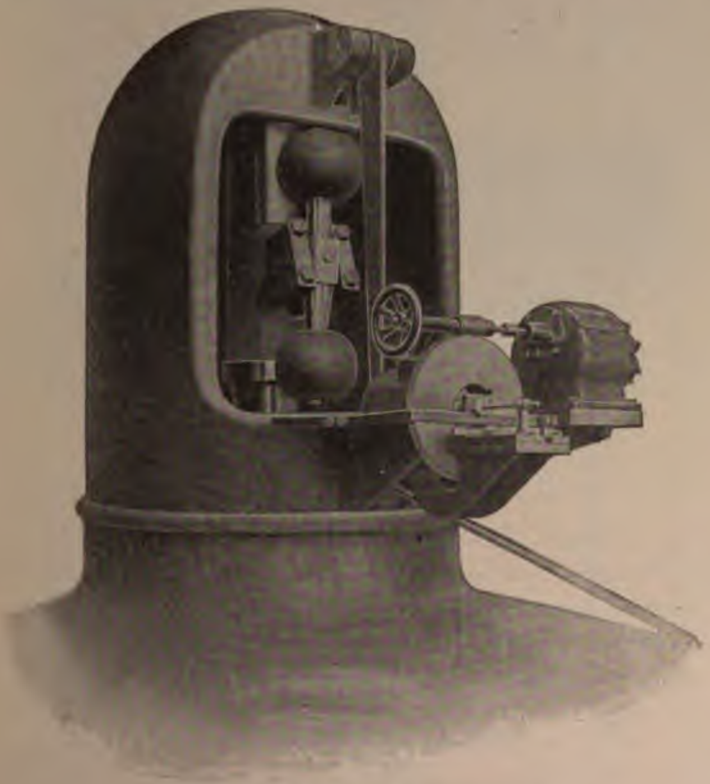

STATIONARY BUCKETS FOR CURTIS STEAM TURBINE.

REVOLVING BUCKETS FOR CURTIS STEAM TURBINE.

The steam consumption of turbines is naturally dependent upon speed, size, and other conditions, and varies in different individual designs. Machines of this type show excellent results as compared with other turbines and engines, the light-load and overload efficiency being a marked advantage. The guarantees of steam economy made by the General Electric Company are based upon results obtained with machines in operation.

Condensers. The larger sizes of Curtis turbines are designed to operate condensing, but they are all adapted to operate noncondensing, and when thus operated will carry full rated load. These turbines are designed to utilize the expansion of the steam to a high degree of vacuum, and the use of good condensing facilities is therefore desirable.

When surface condensers are used, the condensed water can be returned directly to the boilers, as it is entirely free from oil.



GOVERNOR FOR 5000 KW. TURBINE.

Experience shows that distilled water free from air has no bad effect on boilers. The possibility of so returning water is of the greatest practical value, since deterioration of boilers and inefficiency, through dirt and scale, are serious sources of expense in many power stations. In some types of steam turbines, oil is introduced in connection with balancing pistons or steam packing, and therefore this great advantage cannot be realized.

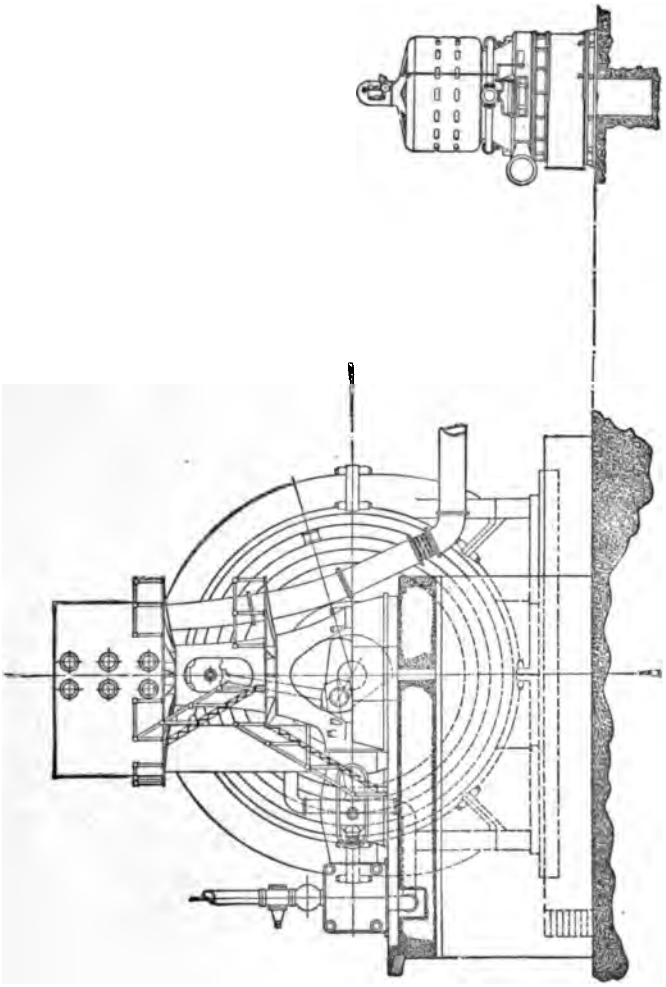
Pressure and Superheat. In these turbines steam is expanded to a moderate degree before it reaches the first buckets. High temperature in the steam is therefore not a source of practical difficulty, and steam of very high pressure and high degree of super-

[illegible] IN COURSE OF CONSTRUCTION

heat [illegible] steam consumption by superheating [illegible] great in the Curtis turbine as in [illegible]

Wear on Buckets. The question is sometimes raised as to the rate of deterioration through erosion of the buckets in the Curtis Steam Turbine. Experiments show conclusively that while the wear varies with different degrees of moisture in the steam and with different degrees of steam pressure, it is in any event negligible from the standpoint of maintenance. All buckets in the Curtis turbine can be renewed without difficulty and at small expense. In the lower pressure stages, where the density of the steam is less, no wear has been perceptible.

Applications. The speeds adopted for the Curtis turbines are such as to give the best results in the design of the generators; consequently the General Electric generators designed for operation with turbines have high efficiency, and are so proportioned that will carry heavy overloads without injurious heating. These nes are built in sizes ranging from 1½ Kw. to 5,000 Kw. The Curtis turbine is also suitable for driving centrifugal

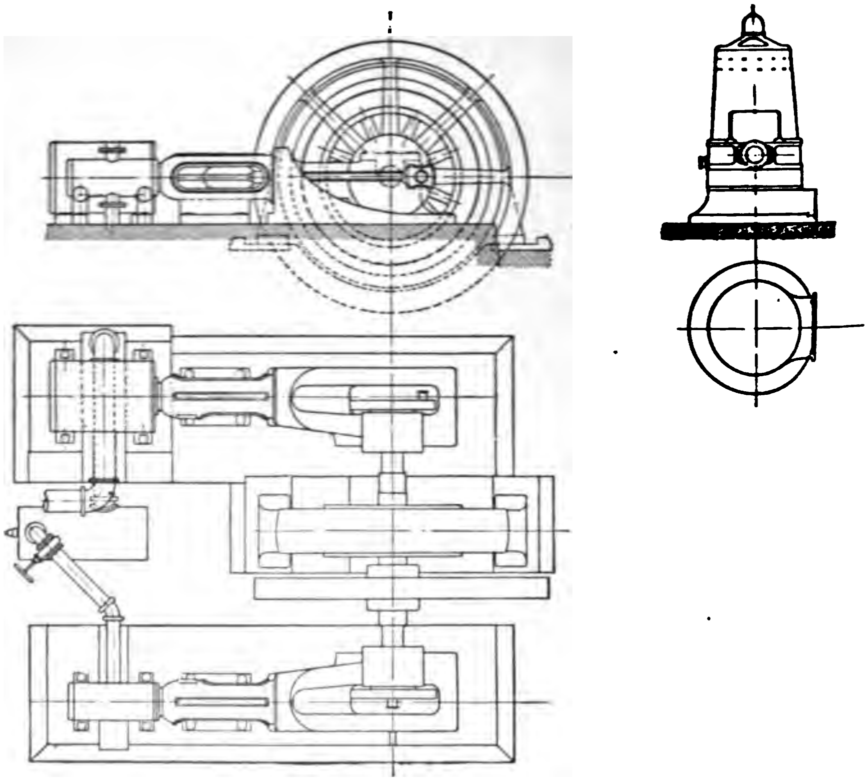


COMPARISON OF 5000 KW. CORLISS ENGINE AND 5000 KW. CURTIS TURBINE.

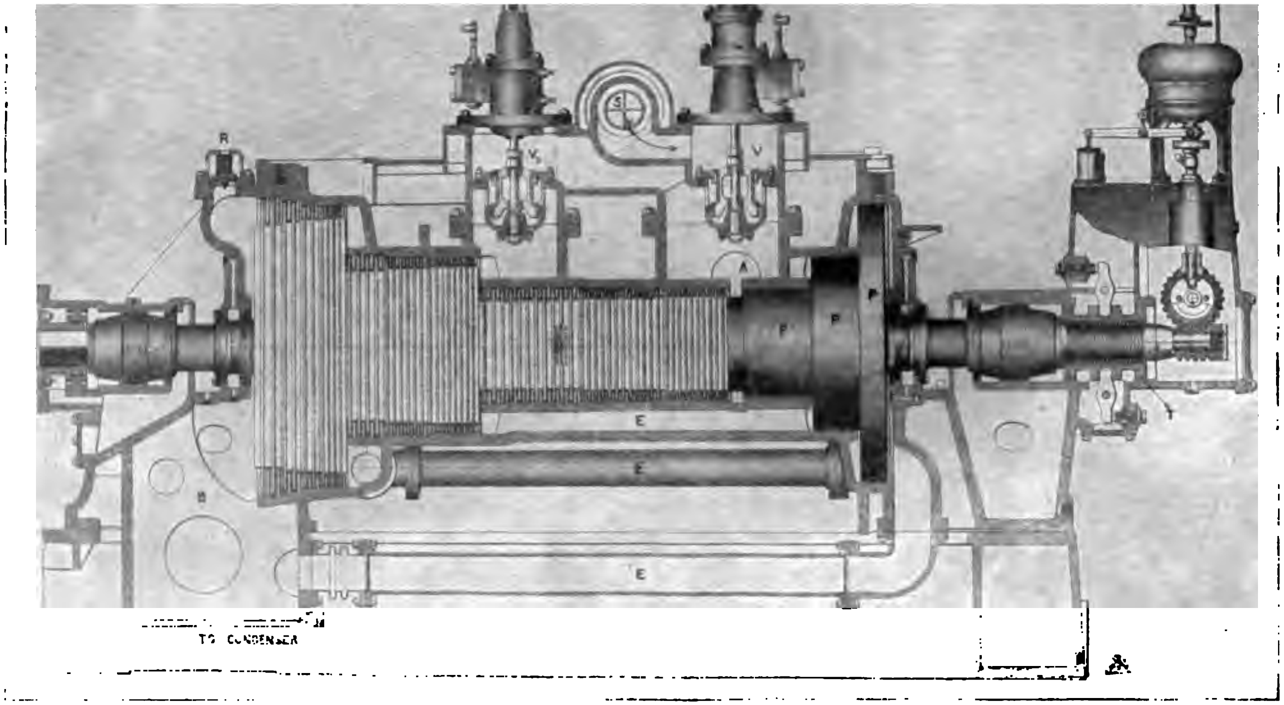
ps, blowers, fans, and other similar apparatus. Turbines for applications are being rapidly developed. In order to meet emand for small direct-current turbines to be used for exciting rain lighting, and isolated lighting, a complete line of non-ensing horizontal shaft turbines has been developed, ranging

in capacity from 1½ Kw. to 300 Kw. These machines are designed to operate at low shaft speed without the use of gearing and show relatively high steam economy when operating non-condensing. They are self-contained and are automatic in regulation.

The machine shown in the accompanying diagram has been tested under a variety of conditions at Newport, and has given results which illustrate very well the advantage of this type. Among other tests that were made, the machine was operated on a rapidly changing railway load; the momentary variations of load amounting to about 120 Kw. In one test the average load carried with this fluctuation was 250 Kw., and the steam consumption was 24.4 pounds per Kw. hour output, with saturated steam. Another



COMPARISON OF 500 KW CROSS-COMPOUND ENGINE AND 500 KW. CURTIS TURBINE.



SECTIONAL ELEVATION OF A TYPICAL WESTINGHOUSE-PARSONS STEAM TURBINE.

test was one with similar fluctuations and with an average of 421 Kw. The steam consumption under this condition was 20.7 pounds of saturated steam per Kw. hour output. The best reciprocating engine under conditions of the first test would probably consume from 28 to 30 pounds per Kw. hour.

The Westinghouse-Parsons Steam Turbine operates on the principle that steam expanding through a definite range of temperature and pressure exerts the same energy whether it issues from a suitable orifice or expands against a receding piston.

Two transformations of energy take place in the steam turbine; first, from thermal to kinetic energy; second, from kinetic energy to useful work. The latter alone presents an analogy to the hydraulic turbine. The radical difference between the two turbines lies in the low density of steam as compared with water, and the wide variation of its volume under varying temperatures and pressures.

A typical Westinghouse-Parsons turbine is shown in section in the illustration.

The steam volume progressively increases from inlet A to exhaust B in the annular space between the rotating spindle and the cylinder walls. The entire expansion, which is approximately adiabatic*, is carried out within this annular compartment which essentially resembles a simple divergent steam nozzle. There is this difference, however, that whereas in a nozzle the heat energy of the working steam is expanded upon itself in producing high velocities of efflux, in the Westinghouse-Parsons turbine the total energy, due to expansion between pressure extremes, is subdivided into a number of steps. In each step the dynamic relationship of jet and vane is such as to secure a comparatively low average velocity from inlet to exhaust, this generally varying from 150 feet per second as a minimum at the high pressure end to about 600 feet per second as a maximum at the low pressure end.

The result is of the utmost importance. With high steam velocities, excessive surface speeds are encountered, causing serious losses from fluid friction and rapid deterioration of parts from erosion. With low steam velocities, commercial speeds are readily obtained, friction loss is greatly reduced, and the depreciation of

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* *i.e.*, No heat is taken in or given out by the steam cycle.

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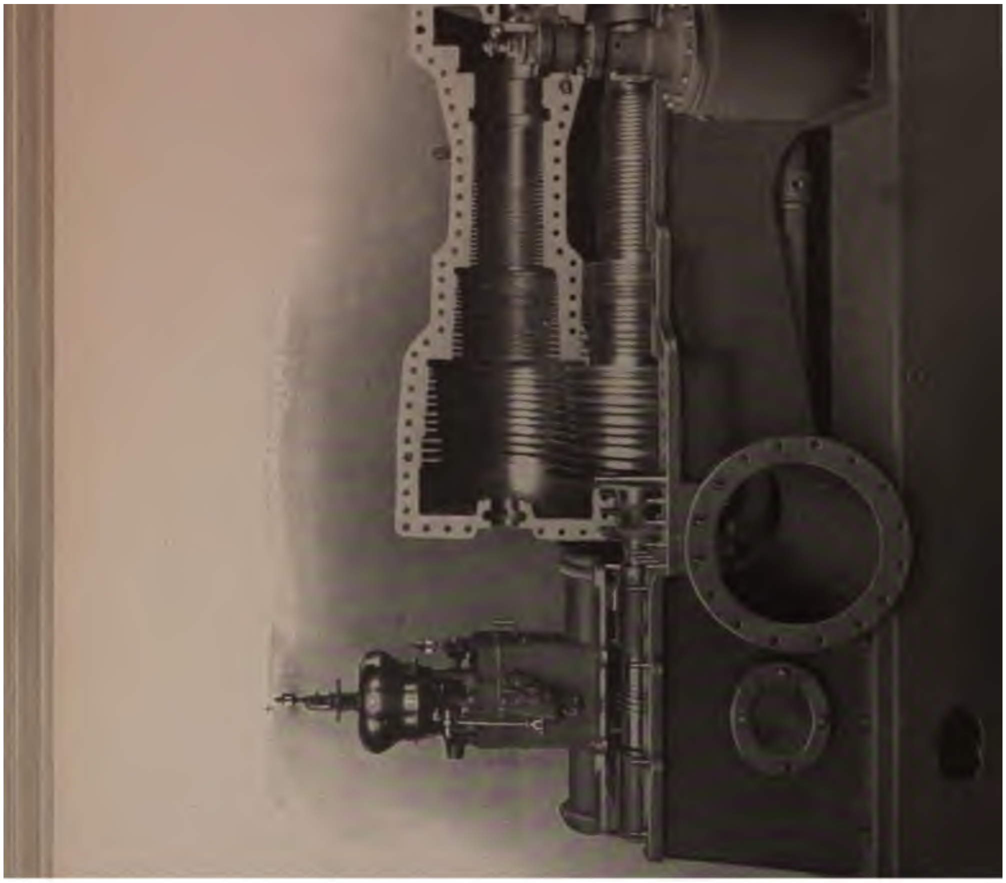


the turbine from practically the only source of wear becomes inappreciable.

The expansion of steam at any one element is typical of its working throughout the turbine. Each element consists of a ring of stationary and a ring of moving blades. The former give direction and velocity to the steam; while the latter immediately convert the energy of velocity into useful torque. The total torque upon the shaft is due both to impulse of steam entering the moving blades and to reaction as it leaves them.

A condensing steam turbine in operation affords a striking example of the conversion of heat into work. The temperature of the cylinder falls, within a distance of three or four feet, from 365 degrees Fahrenheit at the high pressure end to 115 degrees at the exhaust end, when working with 150 pounds of steam pressure and 27 inches vacuum. These temperatures remain constant during operation. There is no alternate condensation and re-evaporation as in the piston engine.

Construction. The Westinghouse-Parsons turbine in effect consists of but two essential elements—a casing, stator or stationary part, and a rotor or rotating part. A brief detailed description follows:

Rotating Element. The rotating element is built up of cast-steel drums carrying rows of blades or vanes, these being mounted on a steel shaft. These drums are arranged in three steps of increasing diameters, but the selection of three diameters is merely for mechanical convenience. Provision for the proper expansion of the steam might be made whether there be one or several diameters. If, however, a speed and diameter of rotor be selected that would permit of a convenient size of blades at the outlet, those at the inlet would become inconveniently small, and *vice versa*. By varying the drum diameters at several convenient points, the proper velocity relations between steam and vane may be preserved, and at the same time the number of different sizes of blades may be reduced to a minimum.

Opposed to the three sets of blades the spindle also carries three rotating balance pistons P, each of such diameter as to exactly balance, by means of the passages E, the axial thrust of the steam against its corresponding drum of blades. These balance pistons [illegible] are not in [illegible] provided with fri [illegible] a path for the stea [illegible] The shaft also [illegible] adjustment bearing T, w [illegible] clearances [illegible] These clearances may [illegible] efficiency. In actual [illegible] and in large blad [illegible]

[illegible] The interior pr [illegible] of the roto [illegible] rings of blad [illegible] revolving blades [illegible] is divided [illegible] the cover all the [illegible]

[illegible] and arrangement [illegible] exhaustive exp [illegible] great ease [illegible]

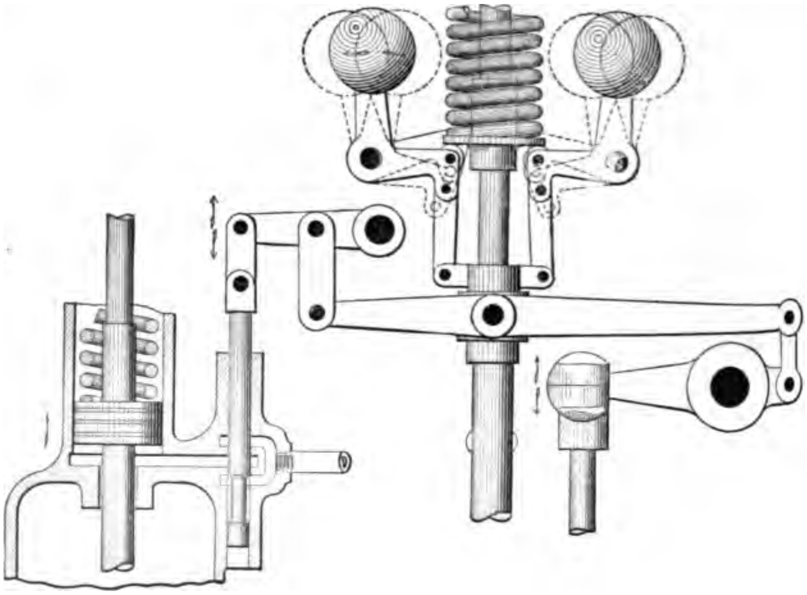
[illegible]

[illegible] bearings are emp

order to permit the spindle to revolve upon its gravity instead of its geometric axis. This expedient is desirable to absorb the vibration which occurs while the turbine is passing its critical speed. The bearing consists of a nest of loosely fitting concentric bronze sleeves with sufficient clearance between them to insure the formation of oil films. These form cushions, permitting a certain amount of vibration of the shaft, but at the same time absorbing and restraining it within narrow limits. In the larger sizes of turbines, however, and in fact for all machines running below 1,200 revolutions per minute, the flexible bearing is not necessary. Instead, a split self-aligning bearing, lined with anti-friction metal, is used as in ordinary forms of moderate speed machinery.

Lubrication. In the Westinghouse-Parsons turbine the bearing surfaces are so liberally proportioned that the entire weight of the rotating element is supported upon a fluid film of oil through capillary action alone, and without the use of oil under high pressures. A small pump driven from a worm gear upon the shaft circulates oil through a closed lubricating system, comprising in the order of their arrangement—pump, oil cooler, bearings and reservoir. The oil is always supplied to the bearings at the point of least pressure; that is, at the top of the shell, from which it is distributed around the shaft. The pressure upon the fluid films is due simply to a static head of one to three feet of oil sufficient to insure thorough flushing of the bearings. It is probable that the shaft never comes in actual contact with the bearings but is separated by the oil film, as is evidenced by the preservation of the original tool marks upon the interior of the shell after years of use.

Governor. Steam enters the turbine in puffs, not in a continuous blast. Speed regulation is, therefore, accomplished by proportioning the duration of the puffs to the load. This is done by means of a small pilot valve actuated directly by the governor and which controls the steam supply through the main poppet admission valve. When the turbine is in operation the main poppet valve is continually opening and closing at uniform intervals, but the periods during which the valve is allowed to remain open are proportioned to the load on the turbine. At light load the valve opens for a very short period and remains closed during the greater part of the interval. As the load increases the period lengthens, until finally, at about full load, the valve does not reach its seat at all and continuous pressure is obtained in the high pressure end of the turbine. On the load becoming further increased an auxiliary or secondary valve begins to open and to admit steam to the annular space at the beginning of the intermediate drum of the rotor where the working steam areas are greater. This increases in proportion the total power of the turbine. The operation of this secondary poppet valve is the same as that of the main admission valve, so that the governor automatically controls the power and speed of the turbine from no load to such overloads as are usually beyond the limits of generating apparatus built on normal ratings. The turbine also operates at its *best economy* at or near full rated load, although possessing at the same time large overload capacity with remarkably high efficiency.



DIAGRAMMATIC ARRANGEMENT OF GOVERNOR MECHANISM.

The governor is of the fly-ball type, the ball levers being mounted on knife edges instead of pins to secure sensitiveness.

The speed of the turbine while running may be varied within the limits of the governor spring by grasping the knurled hand wheel at the top, when the spring and tension nuts may be brought to rest. Adjustment of the spring tension may then be made. This feature is particularly useful for synchronizing the speed of alternating current generators operating in parallel and for distributing the load between them when so operated. The figure shown illustrates diagrammatically the connection between the governor and the pilot valve. Variations in the speed change the height of the fulcrum of the lever on the governor spindle, which in turn varies the throw of the pilot valve relatively to the valve port. This controls the main valve and steam admission as above stated. Reciprocating motion necessary to operate the mechanism originates in an eccentric driven by the turbine from a worm on the main shaft.

Speed Limit. On the larger size turbines an automatic centrifugal speed limit governor is provided which instantly shuts off the steam supply if a predetermined excess of speed above normal should be reached.

Coupling. A flexible sleeve coupling connects the turbine to its generator. Either machine may thus be dismantled without disturbing the remaining adjustments.

The De Laval Steam Turbine, which dates back to 1882, and lays claim to be the pioneer of practical steam turbines, was first constructed by Dr. Gustaf de Laval, of Stockholm, Sweden. His first turbine was of the reaction type and was used in conjunction with his world-famous Cream Separator. In 1888 he built a turbine of the single impulse, free jet type, which differs only in improvement from the engines of the present time. The De Laval turbine has only one set of vanes, and one set of expanding nozzles, in which the complete expansion of the steam takes place in one operation, resulting in a high velocity jet.

The kinetic energy of the jet is successfully utilized by using high vane speeds, easily attained by mounting the turbine wheel, which is of special design, to withstand the centrifugal strains developed at that speed, on a flexible shaft, *i.e.*, a shaft of such dimensions as will allow of the wheel rotating about its center of mass, in place of its center of sphere. This, the "critical speed", usually takes place at about one-sixth of the rated speed. The object attained is that all vibrations due to unequal balancing disappear on reaching this point. It can be noted by anyone who is starting up a De Laval turbine, by the sudden quieting which takes place.

The velocity of the steam jet being given, and with the nozzles at an angle of 20 degrees to the plane of the buckets, the velocity of the turbine wheel should be 47% of the velocity of the steam. The absolute velocity of the steam leaving the buckets is then 34% of the initial velocity, and the formula $\frac{2g \times 550 \times 3{,}600}{V^2 - V_1^2}$ where V and V_1 are, respectively, the initial and final velocities, will give the theoretical steam consumption. With an initial velocity of 4,000 ft. per sec. this would be 9.1 lbs. per horse-power.

Practical considerations, however, confine the De Laval wheel to a lower peripheral speed, *i.e.*, about 1,380 ft. per sec., in place of 1,880, as it should be in the case just given. This gives a theoretical consumption of 9.8 in place of 9.1.

The buckets are separately drop-forged, and are fitted into the wheel by means of a bulb shank fitting into a corresponding slot milled in the wheel.

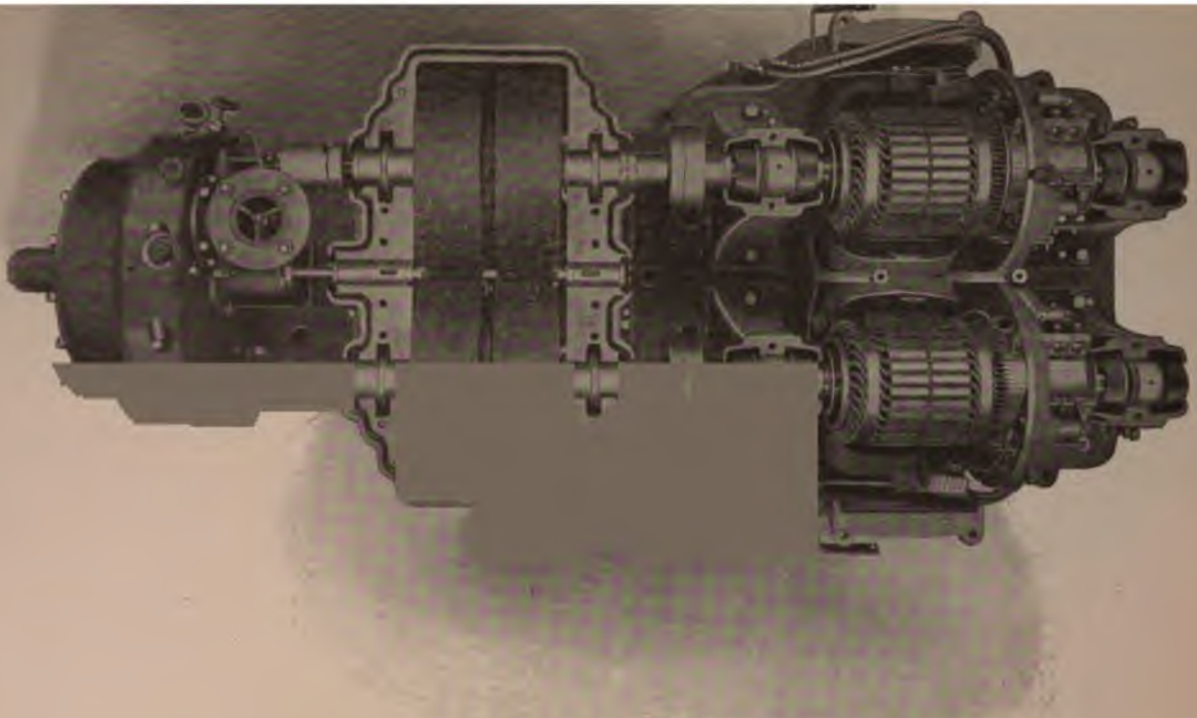
The steam nozzles are of bronze, except where high superheat is used, when nickel steel is substituted. The nozzle section naturally varies with the steaming conditions, *i.e.*, high or low vacuum, and high or low steam pressures, a greater divergence being allowed for the high pressures and high vacuums.

By closing some of these nozzles on a steady light load, the throttling effect of the steam is so decreased as to give excellent efficiencies on light loads.

The gearing is of helical design, as is seen in the accompanying figure, which shows a 110 H.P. turbine dynamo with upper half of gear case and field frame removed.

The high speed shaft bearings are all lined with special high speed metal, and are oiled by gravitation feed, the oil being used over and over in continuous cycle. The low speed bearings are oiled automatically by rings.

The speed governing is performed by a centrifugal governor placed on a low speed shaft and operating on a balanced valve.



110 H.P. TURBINE DYNAMO—UPPER HALF OF GEAR CASE AND FIELD FRAME REMOVED.

In addition to this means of control, an air valve is placed on the wheel case, operated by the governor above a certain speed, admitting air into the wheel case and thereby producing a braking effect on the wheel.

The material used in the construction of the De Laval turbine is only of a high grade. The wheel case is of cast steel; the turbine wheel and shaft of forged nickel steel; the gear wheels of cast iron, with a mild steel rim put on by hydraulic pressure.

The De Laval steam turbine is used on all forms of high speed work, and is especially adaptable for direct connection to electric generators (both alternating and direct current), for centrifugal pumps, and for air or gas blowers.

The De Laval steam turbine will work under all existing conditions from 60 lbs. non-condensing up to 250 lbs. and over, condensing; the nozzles naturally being made to suit the particular conditions.

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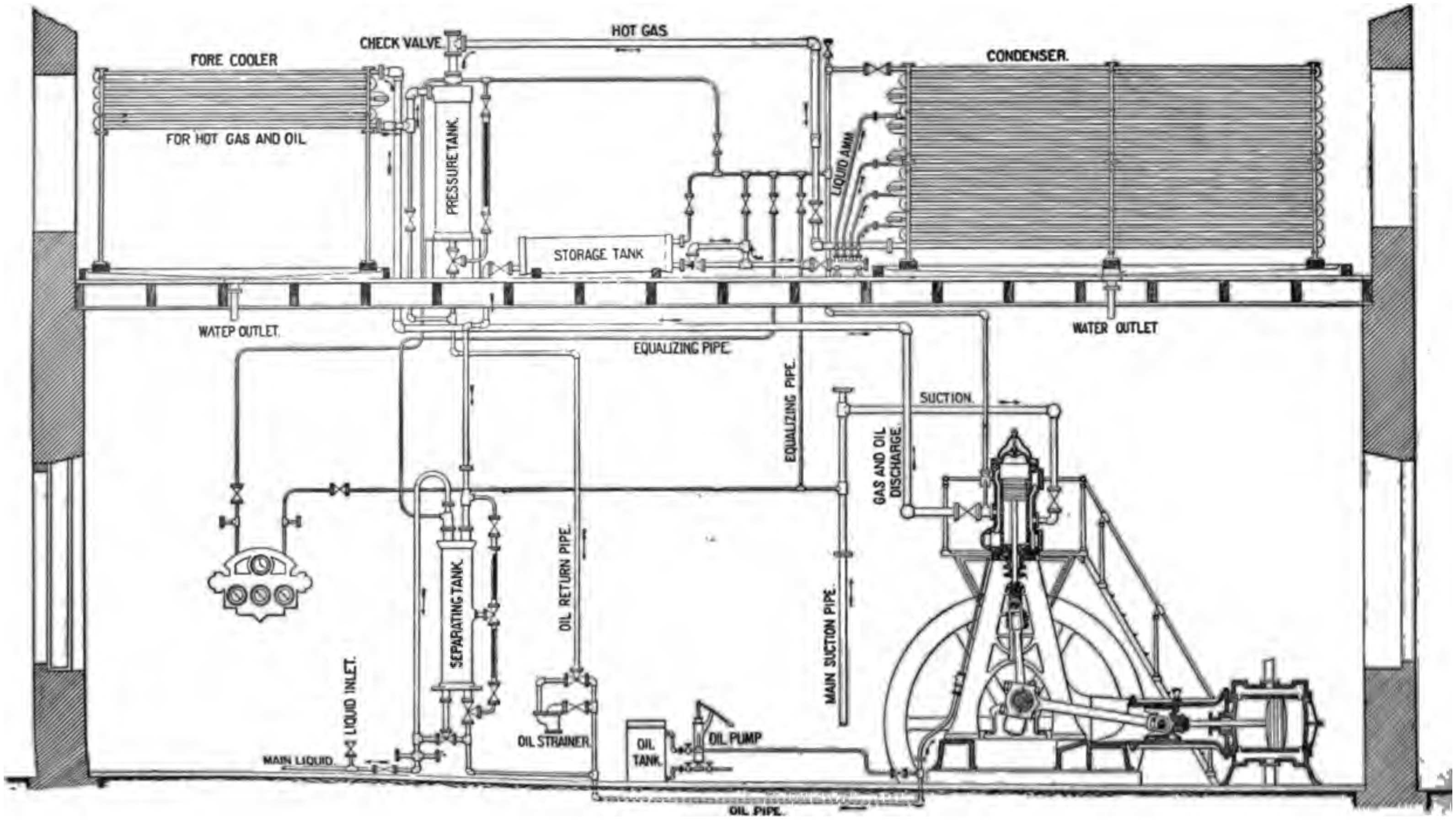

CHECK VALVE
HOT GAS
FORE COOLER
FOR HOT GAS AND OIL
PRESSURE TANK.
CONDENSER.
LIQUID AMM
STORAGE TANK
WATEP OUTLET.
WATER OUTLET
EQUALIZING PIPE.
EQUALIZING PIPE.
SUCTION.
GAS AND OIL DISCHARGE.
SEPARATING TANK.
OIL RETURN PIPE.
MAIN SUCTION PIPE.
LIQUID INLET.
MAIN LIQUID.
OIL STRAINER.
OIL TANK.
OIL PUMP
OIL PIPE.

REFRIGERATION.

Refrigeration may be defined as the process of cooling. It is artificially or mechanically performed by transferring the heat contained in one body to another, thereby producing a condition commonly called cold, but which is in fact an absence of heat. This transfer of heat is most rapidly and economically accomplished by evaporation, but before considering the apparatus used a few important definitions should be reviewed.

Unit of Refrigeration. The unit or basis of measurement of refrigeration, is the unit of heat, and in the United States and England is the British thermal unit (B. T. U.) which is equivalent to raising or lowering the temperature of one pound of water 1° Fahrenheit when at or near 60° Fahrenheit.

Specific Heat. Specific Heat, or capacity for heat, is the relative capacity of a substance for heat, and is stated or expressed relative to that of water, since water has the greatest heat capacity of any known substance except Hydrogen.

Latent Heat. When a body changes from a solid to a liquid, or a liquid to a gaseous state, a certain amount of heat must be supplied to it in order to effect the change. This amount is called its latent heat, and is expressed in thermal units. Thus we have in the melting of one pound of ice a latent heat of 142 thermal units, and we understand by this that in order to melt a pound of ice it must absorb into itself, in making the change, 142 B. T. U., or the equivalent of one pound of water changing 142 degrees Fahrenheit.

Units of Machines or Plants. The unit (or capacity) of refrigerating plants is ordinarily stated in tons, that is, the equivalent of so many tons of ice (of 2000 pounds) at 32° F melted into water at 32° F. The unit is equivalent to 142×2000=284,000 British thermal units.

Thermometers. For ordinary use, a mercury tube having a graduated surface or scale at its back with a bulb at its lower end and containing a quantity of mercury is used to denote the temperature of its surroundings.

Two different scales are commonly used in the refrigerating industries: the Fahrenheit and Reaumer. The Centigrade or French standard is used for chemical or technical purposes. In the United States and England the Fahrenheit scale is generally accepted as standard except in breweries where the German or Reaumer scale is quite often found.

The **Fahrenheit** scale is divided in such manner that the boiling point of water at atmospheric pressure is 212° and the freezing point 32°. It is said that 0° F was the lowest temperature Fahrenheit was able to produce by the melting of ice by salt.

The **Reaumer** scale is graduated by making the boiling point of water 80° while the freezing point is at 0°.

The **Centigrade** has the freezing point of water at 0° as the Reaumer, while the boiling point is fixed at 100°. This graduation is typical of the French system of measurement.

Fig. 1.

To transpose the temperature of one scale to that of the others the table on page 77 will be found convenient.

Let us now take an illustration from a branch of engineering with which almost every one is familiar.

If we place a glass of water in contact with heat at a temperature of 212° F or more, heat will pass from the source and be absorbed by the water until its own temperature reaches that of 212°, after which evaporation of the water commences and continues until the water has all been transformed into steam. During this time an amount of heat corresponding to this duty has been transferred from the source of heat to the water and its vapor called steam. This process is familiar to all engineers, in the steam boiler, and is similar to that of refrigeration as ordinarily applied, except that the one is going on at or above a temperature of 212° F (the boiling point of water) while refrigeration is accomplished by the evaporation of a liquid having a boiling or evaporporating point sufficiently low to obtain the desired temperature.

Agents. Among the most commonly used agents for obtaining artificial refrigeration may be mentioned Ammonia, Carbonic Acid, Sulphur Dioxide and Compressed Air, the first named being the most generally used and approved, while the others have advantages for use on ship-board and other places where the fumes of ammonia would prove objectionable. Ammonia, however, appears to present the most favorable qualities for general use, and will, therefore, be the principal agent considered.

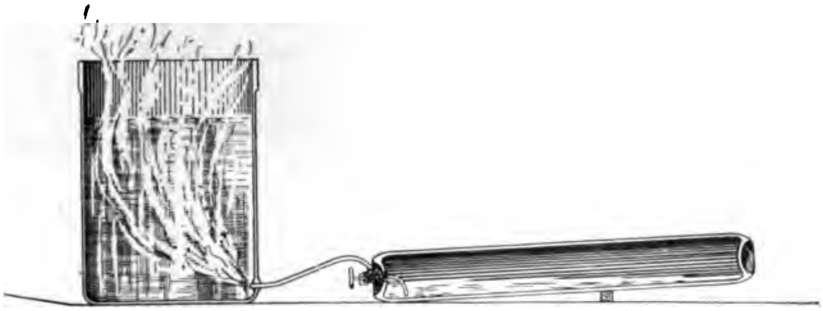


Fig. 2.

Of the two types of ammonia machines, the absorption and the compression, the latter will be the first described. The primitive process of refrigeration is represented by a glass or receptacle (Fig. 1) in which a quantity of anhydrous ammonia is placed, and which, so long as its own temperature and that of its surroundings remain at or above—28° F or 28° below zero (its boiling point) will continue to take heat over to itself, and therefore continue to evaporate and produce a cooling effect upon its surroundings, or what is commonly known as refrigeration in the body or substance with which it is in contact.

In Fig. 2 we have such a receptacle to which is attached a drum or flask filled with the refrigerating agent; if it were possible to procure a cheap volatile liquid, having a sufficiently low

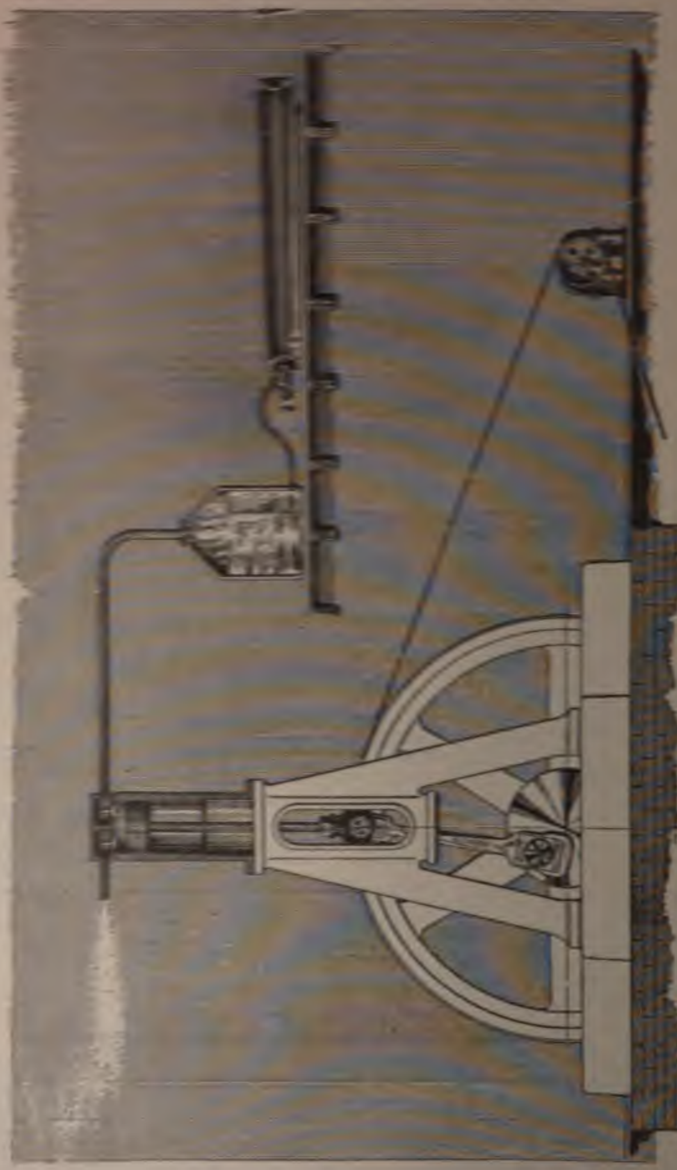


Fig. 3.

boiling or evaporating point, the complex systems of refrigeration would be reduced to the above parts, or equivalent simple system. The system would consist of an evaporator, and a receptacle for the refrigerant, with a connecting pipe between, provided with an expansion valve to regulate the flow of the liquid to the evaporator. The cost of ammonia or the refrigerating agent renders this waste impracticable, at least at the present time; and to make refrigeration a commercial success it becomes necessary to recover and put in condition to be used again the gases arising from the evaporating ammonia. We have seen how the ammonia absorbed a certain amount of heat from its surroundings during evaporation; it is evident that this heat must be taken from it before it can be again effective as a refrigerant, and for this purpose the compressor, condenser and other adjuncts to the plant are required.

The gas pump or compressor is the means employed to recover and compress the gas from the evaporator. In order, therefore, to continue the process of refrigeration in a commercial manner, it becomes necessary to connect to the evaporator and ammonia tank shown in Fig. 2, a gas pump with its engine or other form of power. This is illustrated by Fig. 3. The top of the evaporator is closed and provided with a connecting pipe to the compressor; upon the downward stroke of the compressor piston the gas from the evaporator follows and fills the cylinder above the piston, and upon reaching the bottom of its stroke this valve is closed by a spring, preventing the return of the gas to the evaporator. The return or upward stroke of the piston discharges the gas through the outlet valve and pipe.

Having described the evaporation of the ammonia and recovery of its gases by the compressor, we now supply the apparatus necessary to extract the heat with which it is laden, and thereby cause it to resume its initial state ready to again enter the evaporator and continue the cycle. The apparatus referred to is called the Ammonia Condenser. Fig. 4 illustrates its construction and connection with the balance of the apparatus.

The discharge of the ammonia gas from the compressor is continued through the pipe E, the oil separator F, and into the condenser C, which is composed of a series of pipes over which water flows to take up the heat given out by the compressing of the gas, and which combined effect causes the gas to liquify and flow from the bottom pipe of the condenser through the pipe H to the receiver or storage tank I. This completes the cycle and performs the practical process of refrigeration.

The principle or method of refrigeration has now been described without going into details of construction. The next step is the construction, proportion and combination of the several parts making up the refrigerating plant, and as the evaporator is the foundation or basis of the system, let us first consider this part of the plant.

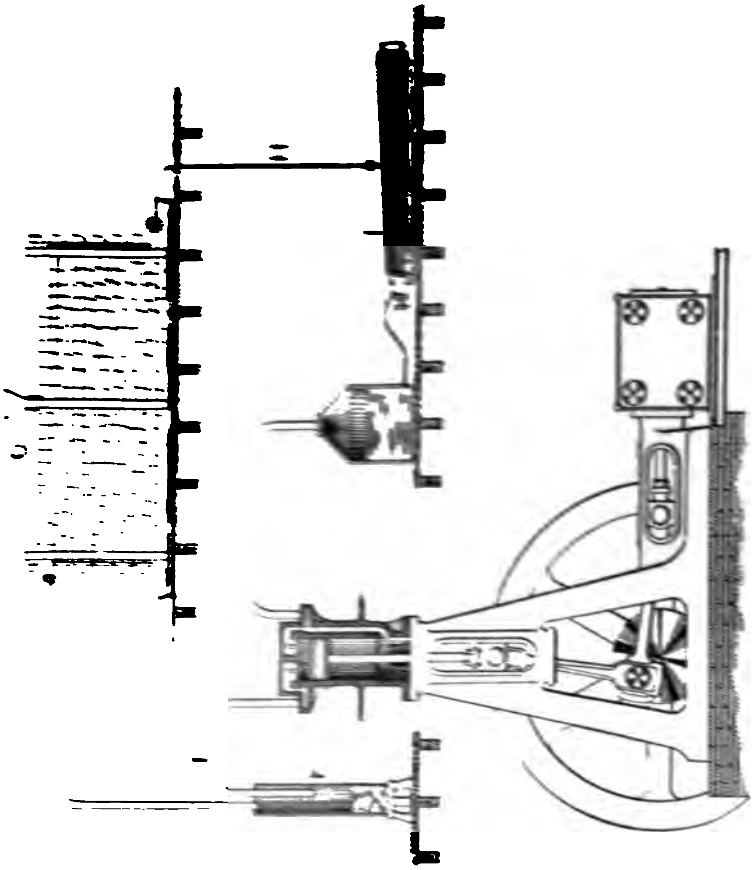
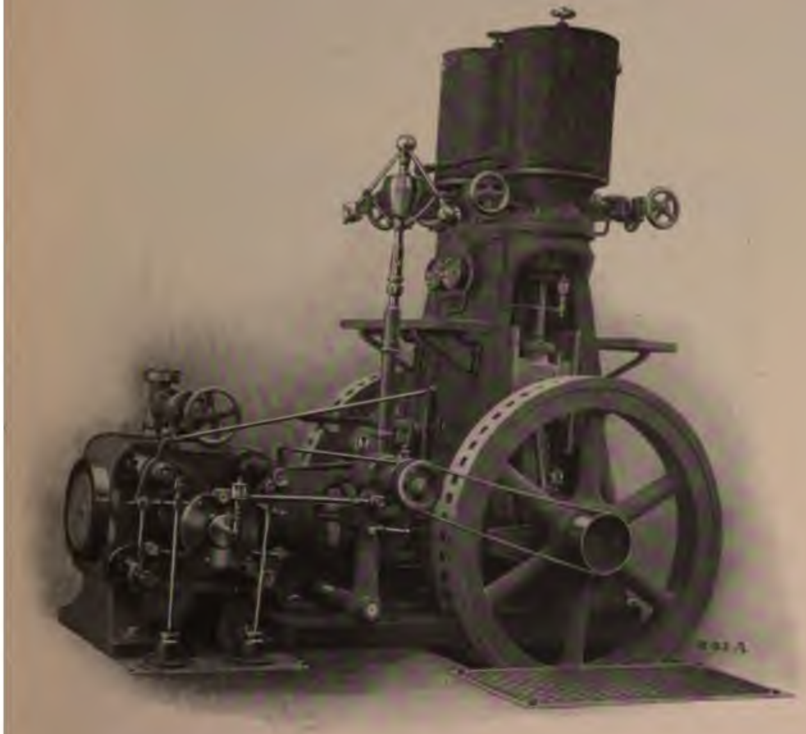


Fig. 4.

EVAPORATORS.

Evaporators may be divided into two classes: The first is



15 TO 20 TON REFRIGERATING MACHINE.
Frick Company.

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operated in connection with the brine system. In this evaporator salt brine (or other solution) is reduced in temperature by the evaporation of the ammonia or other refrigerant and the cooled brine circulated through the room or other points to be refrigerated. In the second, the direct-expansion system, the ammonia or refrigerant is taken directly to the point to be cooled, and there evaporated in pipes or other receptacles, in direct contact with the object to be cooled. Which of the two systems is better, is a much disputed and debated point; we can state, in a general way, that both have their advantages, and each is adapted to certain classes of duty.

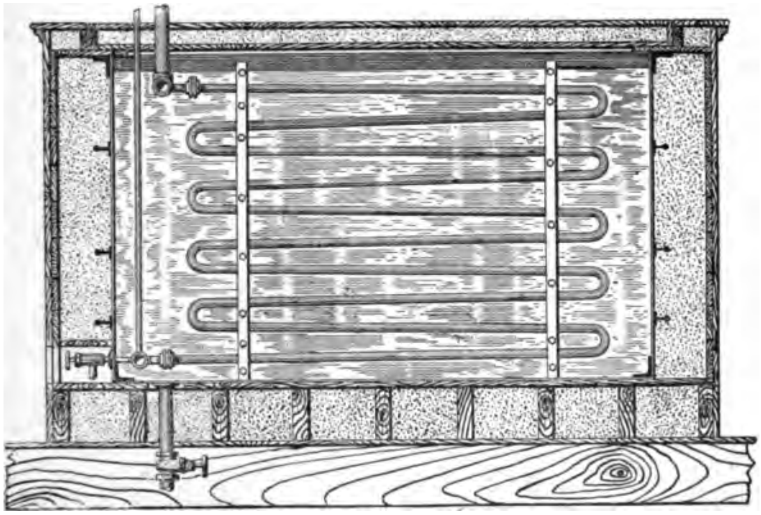


Fig. 5.

The cooling of brine in a tank by a series of evaporating coils, (one of the earliest methods) is common to-day. A description of the many methods of construction and equipment would require much space. Let us, therefore, discuss the two most general types only, viz: the rectangular with flat coils, and the circular with spiral coils.

Fig. 5 shows a sectional view of a brine tank. Flat or zigzag evaporating coils are connected to manifolds or headers; the pipe connections leading to and from these manifolds for the proper supplying of the liquid ammonia and the taking away or return of the gas to the compressor are also shown. For coils of this type. 1-inch or 1¼-inch pipe is preferable, owing to the impossibility of bending larger sizes to a small enough radius to get the required amount into a tank of reasonable dimensions. It is possible to make coils of this construction of any desired length or number of pipes to the coil, "pipes high," the bends being from 3½ inches to 4 inches centers for 1-inch pipe, and 4 inches to 5 inches for 1¼-inch pipe. It is preferable to make the coils of moderate length, (not less than 150 feet in each) and there is no disadvantage (other than in handling) in making them to contain up to 500 or 600 feet each. It will be observed that there is a slight downward pitch to the pipes with a purge valve at the lowest point of the bottom manifold, which is, undoubtedly, valuable and an almost necessary provision. This valve is for removing foreign matter, which may enter the pipes at any time, and by opening the valve and drawing a portion of their contents, the condition of cleanliness can be determined without the necessity of shutting down and removing the brine and ammonia for inspection. The coils are usually strapped or bound with flat bar iron about ¼ inch × 2 inches (or a little heavier for the longer coils) and bolted together with ½-inch square-head machine bolts The coils are painted with some good water-proof or iron paint.

The brine tank is usually constructed of iron or steel plates, varying from [illegible] inch to ⅜ inch in thickness; the average being ¼ inch for tanks of ordinary size. The workmanship and material for a tank for this purpose should be of the very best; without these the result is almost certain to be disastrous to the owner or builder. The general opinion with iron workers (before they have had experience) is that it is a simple matter to make a tank which will hold water or brine, and that any kind of seam or workmanship will be good enough for the purpose. On the contrary the greatest care and attention to detail is necessary. It is customary, and good practice, to form the two side edges at the bottom by bending the sheets, thereby avoiding seams on two sides, while for the ends an angle iron may be bent to conform to this shape and the two sheets then riveted to the flanges of the angle iron. The edges of the sheets should be sheared or planed bevel, and after riveting, calked inside and out with a round-nosed calking tool. The rivets should be of full size, as specified for boiler construction and of length sufficient to form a full conical head of height equal to the diameter of the rivet and brought well down onto the sheet at its edges. An angle iron of about 3 inches should be placed around the top edge, and riveted to the side at about 12-inch centers.

One or more braces (depending on the depth) should extend around the tank between the top and bottom, to prevent bulging; without these it would be impossible to make the tank remain tight, as a constant strain is on all its seams. A very good brace for the purpose is a deck beam. Flat bar iron placed edge-wise against the tank with an angle iron on each side and all riveted through and to the side of the tank with splice plates at the corners or one of each pair long enough to lap over the other makes a good brace; heavy T-iron is also used to some extent. It is usual to rivet up the bottom of the tank and a short distance up the sides, then test by filling with water; if tight lower to its foundation and complete the riveting and calking. It may then be filled with water and tested until proven absolutely tight, when it may be painted with some good iron paint; it is now ready for its equipment of coils and insulation.

A washout opening with stop valve should be placed in the bottom at one corner; for this purpose it is well to have a wrought iron flange, tapped for the size of pipe required and riveted to the outside of the bottom. If the brine pump can be located at this time, it is well to have a similar flange for the suction pipe riveted to the side or bottom of the tank, as a bolted flange with a gasket is never as durable as a flange put on in this manner.

Assuming that the tank is now absolutely tight and painted, the insulation may be put around it, the insulated base or foundation having been put in previous to the arrival of the tank. The insulation should be constructed of joists 2 inches or 3 inches × 12 inches on edge and filled in with any good insulating material and floored over with two thicknesses tongued and grooved flooring with paper between. In putting the insulation on the sides and ends of the tank, place joists 3 inches × 4 inches resting on the projecting edges of the foundation about 2 feet apart. The upper ends should be secured to the angle iron at the top of the tank, its upper flange having been punched with $\frac{5}{8}$-inch holes 18 inches to 24 inches centers and to which it is well to bolt a plank, having its edge project the required distance to receive the uprights. Between the braces around the tank blockings should be fixed to secure the frame work at the middle, as the height of some tanks is too great to depend on the support at top and bottom alone.

After the frame work has been properly formed and secured to the base and tank, take 1-inch flooring, rough, or planed on one side, and board up on the outside of the uprights, filling in as the work progresses with the insulating material which may be any one of the usual materials, granulated cork being about the best, all things considered, although charcoal, dry shavings, saw-dust, or other non-conductors may be used with good results. When the first course of boards is in place it is well to tack one or two thicknesses of good insulating paper against the outer surface, care being taken that the joints lap well and that bottoms and corners are filled and turned under at the junction with the bottom insulation. It is then in shape for the final or outer course which is very often made of some of the hard woods in [illegible] [illegible] widths, tongued, grooved and beaded and finished off with a base board at the bottom and moulding at the top, and given a hard wood finish in oil or varnish. If the tank is located in a part of the building in which appearance is of no importance, the outer course may be a repetition of the first, except that the boards are put on vertically instead of horizontally.

It is well to make the top of the tank in removable sections to facilitate examination or cleaning; for this purpose make a number of sections about 2½ to 3 feet wide of the length or width of the tank, using joists about 2 inches × 6 inches placed on edge, floored over top and bottom and filled in with the selected insulating material. It is also well to have a small lid at one end of each (preferably over the headers or manifolds) which will allow of internal examination of the tank to ascertain the height or strength of brine without removing the larger sections. The tank is now fully equipped and ready for testing and filling with brine.

For a circular tank the general instructions regarding construction and insulation may apply as with the rectangular tank just described; therefore only its special features will be considered. If the tank is small and there is sufficient head room above it for handling the coils there cannot be serious objection to this type as its cost is lower than that of the rectangular tank. This is often an important item in a small installation, but when the tank is of considerable size and the coils large it is not as readily

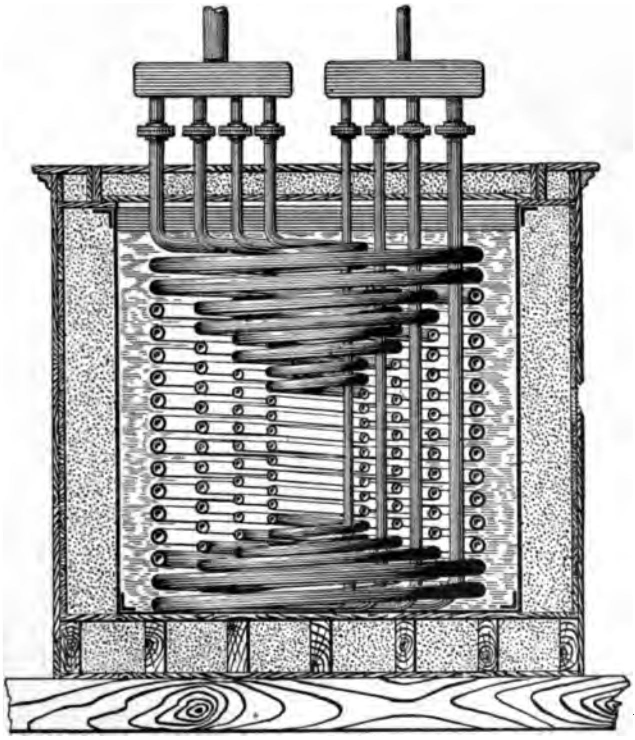


Fig. 6.

handled and taken care of as the other type. The usual construction of a nest of coils for a round tank is to bend the inside coil to as small a circle as possible, which, if it be of 1-inch or $1\frac{1}{4}$-inch pipe may be 6 to 8 inches. Increase each successive coil enough to pass over the next smaller until the required amount of pipe is obtained. The ends may then be bent up or out and joined to headers at top and bottom and the tank insulated in the manner previously described; it is then ready to test and charge with ammonia and brine. Fig. 6 represents an evaporator of this type.

Other constructions of tanks and coils are too numerous to describe in detail, and with one exception may be properly classed in one of the preceding types. The one exception referred to is illustrated in Fig. 7. It is quite common and is adopted for large pipe and is often called oval, although not of that shape, but rather a combination of the flat and circular form. It has some good features; it allows the maximum amount of pipe in the smallest space and a large amount of pipe in a single coil.

The Brine Cooler at present is a popular and efficient method of cooling brine for general purposes. Owing to mechanical defects and the impossibility of obtaining a brine solution which would not freeze, it was abandoned only to be taken up again, and with the aid of modern ideas and better material it has become highly successful. Unlike the brine tank and coil method of refrigeration, in which the ammonia is evaporated within the coils while the brine surrounds them, the brine passes through a series of coils and the ammonia evaporates within a wrought or cast shell surrounding the coils. Thus the action is reversed.

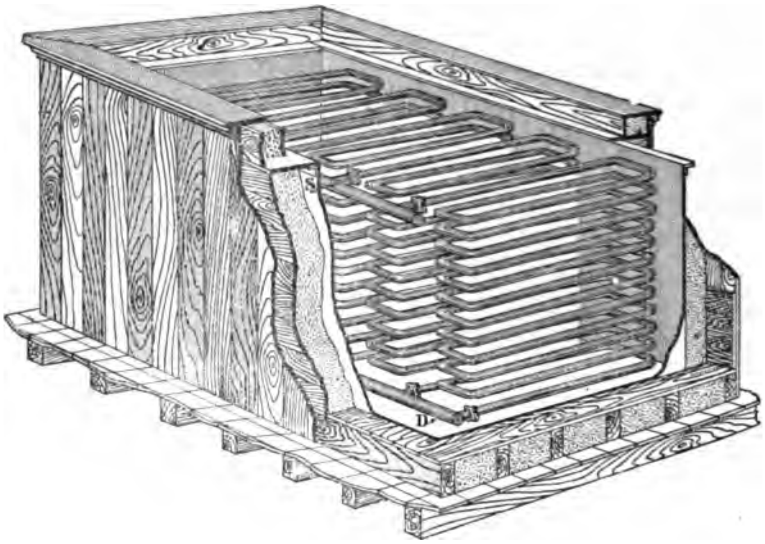


Fig. 7.

Fig. 8 is what is known as the **enclosed-shell type** of brine cooler, A representing a cast or wrought-iron shell, flanged at each end, to which heads are bolted; a tongue and grooved joint between, makes the closed cylinder gas tight. B is a series of welded extra strong pipe coils, one within the other, having their ends project through the heads and joined to headers at each end with proper unions; a stop valve is supplied to each coil. Lock nuts or glands are placed around each coil at its opening through the heads and one or more glass gauges with the usual gauge fittings are tapped into the side of the shell to indicate the amount of ammonia. Liquid ammonia is fed into the shell by an ordinary expansion or feed valve near the lower end of the shell at C and the gas taken off through the opening and pipe D to the compressor. A purge valve E for drawing off impurities is placed at the lowest point in the bottom head and a second one F for gas or air in the top head.

In operation, the discharge pipe from the brine pump is connected to the top header or manifold, and the bottom header is connected to the main leading to the refrigeration to be performed; the return from the cooling system is generally brought to a medium-sized brine tank without coils to which the suction of the pump is connected, thereby completing the brine circuit. The heat of the brine passing through the coils in the cooler is taken up by the ammonia during evaporation; the

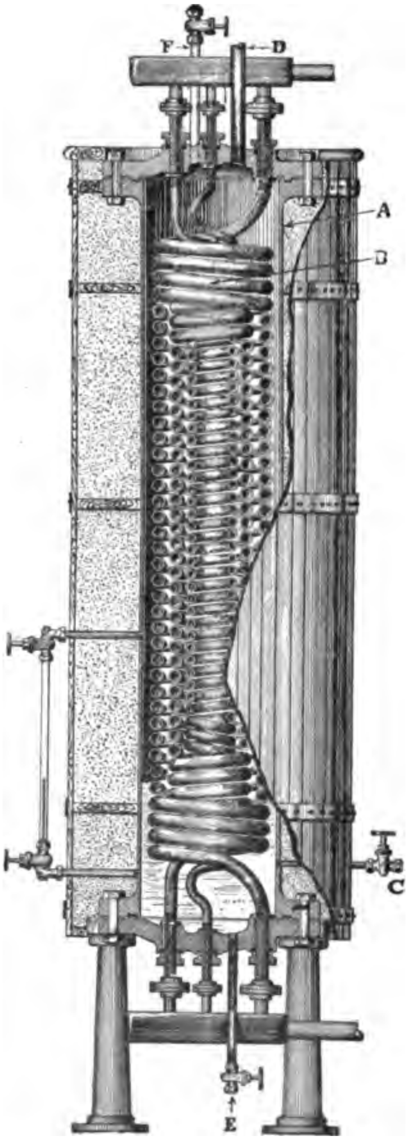


Fig. 8.

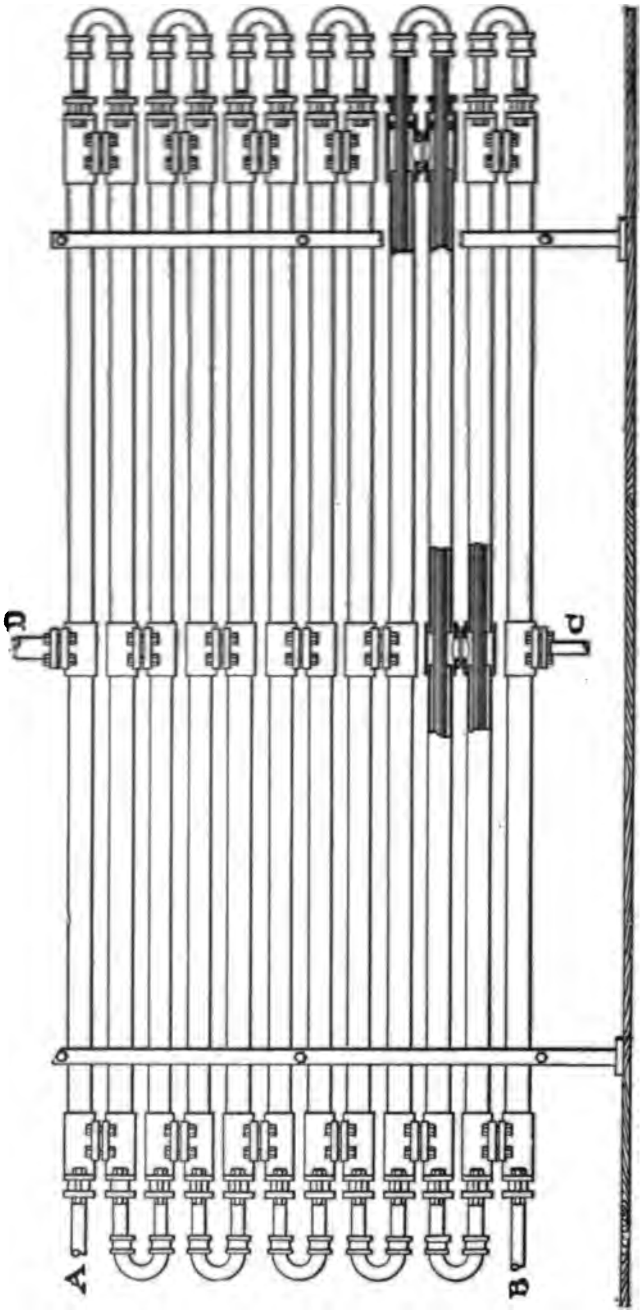


Fig. 9.

gas is taken away from the top of the shell to the compressor, and discharged to the condenser and into the receiver to be re-used, as with the system illustrated by Fig. 4.

The advantages of the brine cooler over the brine tank are due to two features of construction. The brine in passing through the coils is divided into a number of small streams, and while flowing rapidly through a coil of considerable length is churned to such an extent that it is all brought in contact with the coil, of which the outer surface is exposed to the temperature of the evaporating or boiling ammonia, and thereby gives up its heat much more rapidly to the ammonia than in the brine tank. In the brine tank there is a large body of brine with scarcely any movement; the brine immediately in contact with the pipe may be quite cold, but becomes warmer as the distance from the pipe increases.

The second advantage is that during the evaporation of the ammonia a violent ebullition is taking place, and if this is confined within the pipe coil a certain amount of the liquid is carried forward by the escaping gas, and the evaporation within the coil is limited. This limit is reached when the vapor enters the compressor in such quantities as to cause too great an expansion in the compressor, or when it becomes difficult to keep the stuffing-box tight, while in the brine cooler properly constructed, the larger diameter of shell allows evaporation to take place place up to practically the theoretical boiling point of the ammonia, without the liquid being carried forward through the gas pipe to the compressor. This permits a much higher evaporating pressure or "back pressure" as it is commonly called, than with the brine tank and coils. Also the concentrated or compact shape and construction of the cooler allows greater economy than the large radiating surface of the brine tank. The sides and end of the cooler are insulated by some one of the usual methods and very often lagged with finished hardwood strips, tongue, grooved and beaded and bound with finished brass or nickel plated bands.

Salt brine is commonly used in the brine tank and coil form of refrigerator, but it is unsafe to use it with the brine cooler because when passing through the coil it may freeze and burst the coil. A solution of chloride of calcium is commonly used, its freezing point being 54° below zero Fahr. while salt brine is practically 0° Fahr.

The second form of brine cooler is what is known as the **double-pipe type,** the construction of which is illustrated in Fig. 9, In this type one pipe is within another, the brine being discharged from the pump into one or more pipes at A and issuing at B. This connection leads from the main to the point to be refrigerated and the ammonia is expanded or fed into the annular space between the two pipes, and takes up the heat of the brine in evaporating and issuing as gas from the opening D at the top of the cooler. From thence the ammonia passes to the compressor and through the cycle of compression, condensation and return to the liquid ammonia receiver as before. The ammonia evaporating between the two pipes will naturally absorb as much heat from the outside surface as from the inner or brine if allowed to do so, and it therefore becomes necessary to insulate the outside of this from exterior influences. As it is practically impossible to cover the bends and irregular surfaces it becomes necessary to build an insulated room in which the cooler is erected. This is commonly done, but some authorities do not consider it the most practical form. The cooler is equipped with the necessary stands, manifolds and stop valves to properly control the action of any one section when more than one is used.

As with the enclosed brine cooler, previously described, chloride of calcium brine should be used, as there exists the same liability to freeze, and it is unsafe to operate either type with the ordinary salt solution. These two types of brine coolers and brine tanks and coils are the usual means of cooling or refrigerating brine.

COMPRESSORS.

The next step in the process is the recovering of the evaporate ammonia and its return in liquid form. Compressors may be divided into two principal classes, single acting and double acting; and each class subdivided into vertical and horizontal. They are of the vertical type if single acting, and of both vertical and horizontal if double acting, although the majority of the double acting are of the horizontal type. Machines may also be classed according to the form of driving. The engines used may be horiontal or vertical, and within this classification comes almost any achine of modern build.

Fig. 10 illustrates the vertical single-acting type of compressor, two in number, in combination with and driven by a orizontal Corliss engine. It embodies the necessary and usual equisites of an efficient gas pump or compressor.

As already stated, the function of the compressor is to ecover the gas from the evaporator and compress it into the conlenser at a pressure which will cause it to liquify under the ction of the cooling water. It is evident that the gas must ollow the piston in its downward stroke and fill the compressor, nd upon its reaching the end of its stroke and the pressure being balanced, the strength of the spring in the suction valve causes it to close and the piston begins its return stroke. The compression f the gas within the compressor takes place until its pressure quals or slightly exceeds that above the discharge valve; it then pens and the compressed gas flows into the discharge pipe nd thence to the condenser. Two compressors of the singlecting type are almost always used in a machine of this type, and the cranks set opposite, or at 90° to one another, so that one compressor is filling and one compressing and discharging at each half revolution of the crank shaft, and the load is accordingly divided into two units, either one of which may be operated independently if necessary.

As the evaporator is the heart of the refrigerating system, so the piston and valves are the heart of the compressor, and the principal, almost the only, cause of difficulty in the action of the compressor will be found to be in one of the two.

In the compressor in which the valves operate in a cage, there must of necessity be a gas-tight joint between the bottom of this cage and the compressor head or piston in which it is located; this joint is made in a variety of ways, any one of which will prove effective. A square shoulder is cut into the head with a orresponding shoulder on the cage to match, and a lead gasket bout $\frac{1}{16}$ to $\frac{1}{8}$ inch in thickness placed between the two. This makes a durable joint, except in cases where the joint between the two is of such an amount that the lead is constantly pressed through (a disadvantage of lead as a gasket material). Without

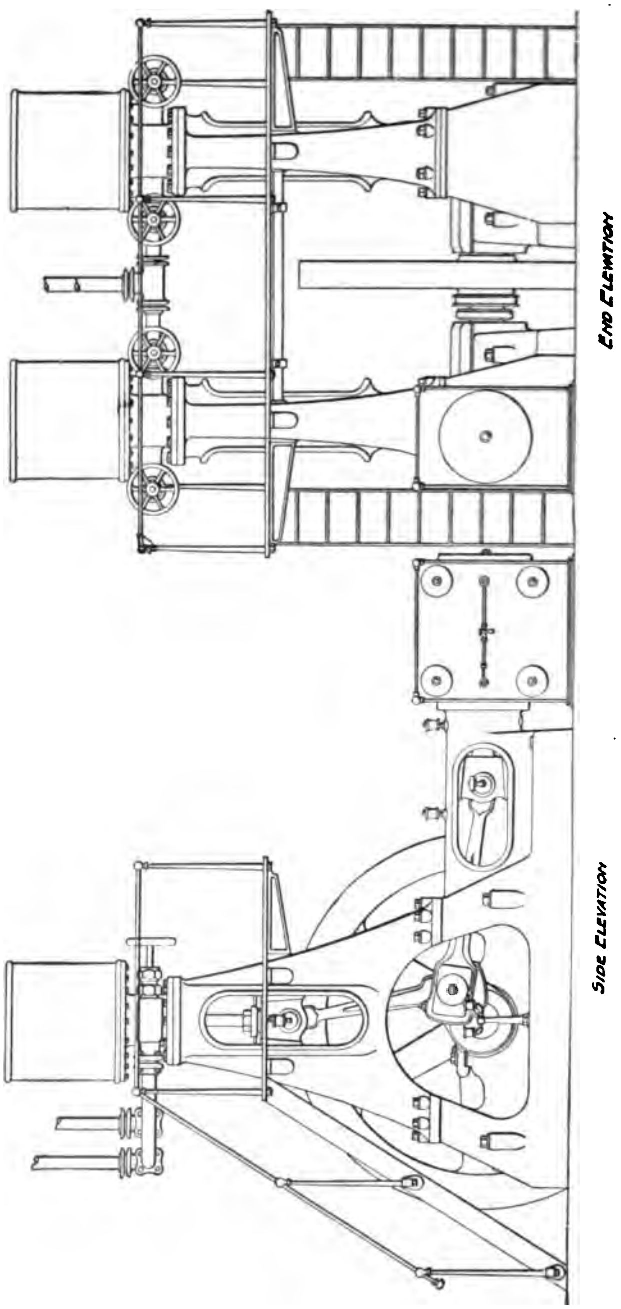


Fig. 10.

one's being aware of the fact, the gasket is gone and a leakage exists. Another objection to a lead gasket is that it compresses and *knits* into the interstices between the cage and the head. This often makes it impossible to remove the valve or cage from the head without the aid of some kind of a chain block or tackle. Another form of gasket for this purpose, but not popular from a lack of confidence in its permanency, is common lamp or candle wicking saturated with oil and wound tightly and smoothly in the corner against the shoulder on the cage. It is put in place and fully compressed by pulling down on the valve cap until the cage is at the desired height. It is good practice to wind enough on until, when the cage is pressed down by hand, it stands about $\frac{1}{16}$ inch above the surface of the head at the top; this amount of compression would make it about even when pulled down. As already stated, this kind of packing is not very popular on account of lack of durability, but it gives very satisfactory results when properly applied, and the valve and cage may be easily removed. Recently the gasket has been omitted and a ground joint made between the cage and the head. Although its efficiency has not yet been proven, from present indications it should be both satisfactory and permanent.

Assuming that we have made the joint between the cage and the compressor head gas-tight, we must also be certain that the compressor valve forms a perfect joint in closing against the cage. In a vertical compressor, in which the action of the valves is also vertical within their cages, the conditions naturally favor this to the greatest possible extent, as in closing they drop to their seats and it is only necessary to provide for the slight wear taking place in the stems and on the seats due to the rapid opening and closing of the valves. The valve stem must necessarily fit the guides in the cage as closely as possible and still allow free movement, and the seat between the valve disc and cage (preferably made at an angle of 45°) must first be machined to the proper angle and then ground in the usual manner. Having made it impossible for the gas to pass the valves and cages, means must be provided for closing the valves at the proper time to prevent loss, as a valve which is slow in reaching its seat presents a double evil—loss of efficiency and improper or irregular action on the balance of the machine. The spring is the usual means of opening the compressor valve, and, on the suction or inlet, is commonly placed between the top of the cage and underneath a washer placed at the end of the stem and held in place with one or two nuts; it is frequently reinforced with a buffer spring, or one of greater strength, enough shorter than the length of the stems to allow its opening the proper distance, and then stopping the travel of the valve gently.

Valves. Fig. 11 illustrates a type of inlet valve which has stood the test for years and embodies many good qualities. The different points referred to are indicated by the following letters: A, the joint between the cage and head; B, the contact between the valve disc and the cage; C, the spring for actuating the valve, and D, the buffer or stop spring to stop the travel of the valve at the proper point of opening. In the use of this valve the requirements are that it shall admit the gas to the compressor during the downward stroke of the piston, and close while the piston is at the bottom of stroke and the crank pin is passing the bottom center. It will, therefore, be readily understood that if the spring is stronger than necessary, it will require a certain amount of the pressure of the gas to overcome the strength of the spring, and prevent the filling of the compressor to its fullest limit. In the closing of the valve it would be driven with considerable force against its seat, making a noise in so doing and causing excessive wear. Considerable skill and experience are required to obtain perfect results, but with good judgment and a few trials most of the imperfections can be overcome. Generally the closing spring should not be stronger than is necessary to close the valve when held vertically in the position in which it naturally rests. By taking the valve and cages in the hands and pressing down on the top of the stem with one of the fingers, it will be readily ascertained when the springs are of proper strength to close the valve. When put on the compressor, so far as the operation of the inlet valves is concerned, the machine will be practically noiseless and effective in the admission and retention of the gas from that side.

The discharge valve operates in the reverse direction to that of the suction valve. We know that the suction valve closes while the compressor piston is at its lowest position in the compressor and while the crank is passing its bottom center. The piston now moves upward compressing the gas until it reaches the pressure in the ammonia condenser, and as it passes this point far enough to overcome the tension of the spring on the discharge valve, it causes it to lift and the contents of the cylinder in its compressed state are discharged through the discharge pipe into the condenser. This continues until the piston reaches its highest point and the crank is passing its top center, at which point the valve closes and the suction valve again opens to admit an additional amount of the gas. This process is continued during the operation of the apparatus.

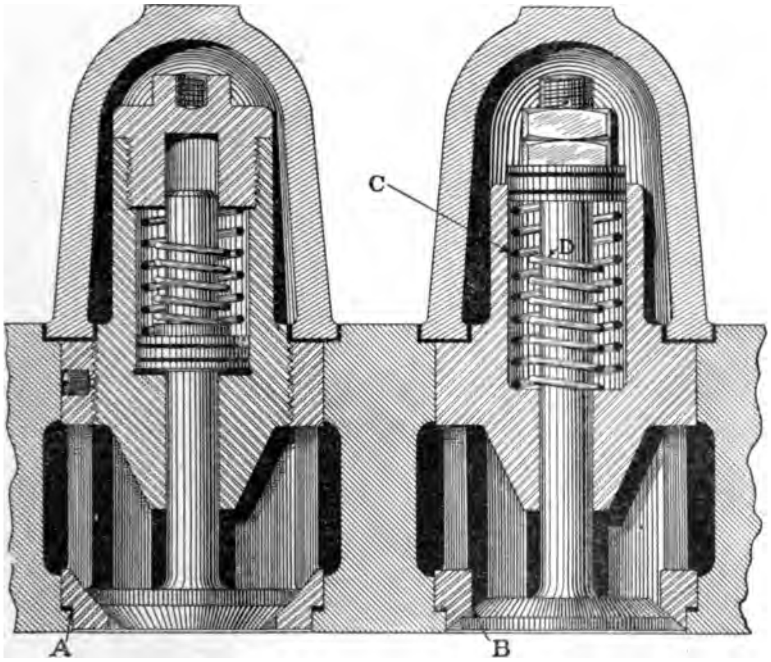


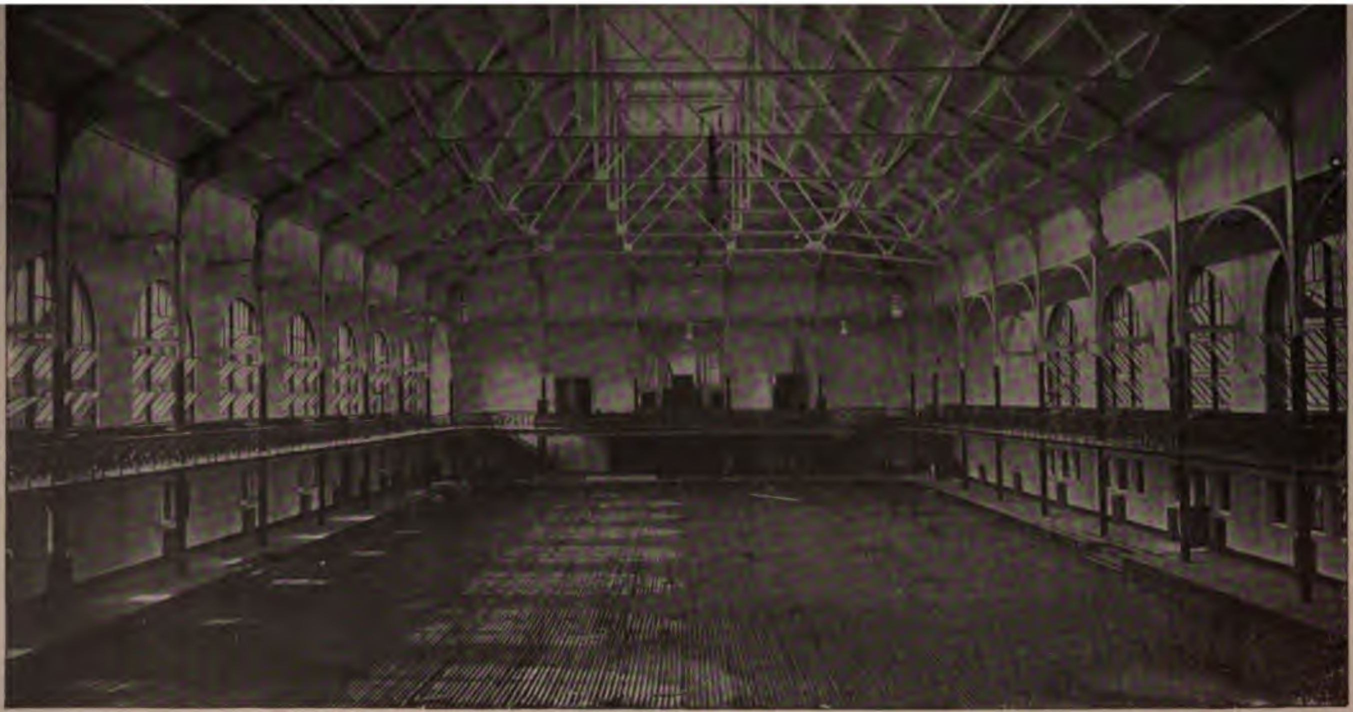
Fig. 11.

It will be noticed that the suction valve opens and closes while the piston is practically without motion, but the discharge valve opens while the piston is nearly at its maximum, and closes while the piston is at the minimum speed. From this it will be apparent that the thrust or effort on the discharge valve is much greater than on the inlet, that is, in an upward or outward direction; hence the device for arresting the upward motion of the valve must be more effective than the other. Also the valve must close in a shorter space of time because the great pressure above the valve would cause it to fall with considerable force were it not to reach its seat while the piston was still at the top of its travel, and the pressure above and below equal. Should the piston begin its return or downward stroke before the valve closes it would have the condensing pressure above, and the inlet pressure below, a difference of from 150 to 175 pounds. This pressure would cause it to seat with an excessive blow which would soon cause its destruction, and also cause the escape of a portion of the compressed gas into the compressor resulting in great loss in efficiency of the machine.

To stop the valve in its opening and prevent shock, it is necessary to provide a buffer or cushion in addition to the spring used to close it. The cushion, as usually constructed, is a piston fitting closely in a cylinder and provided with openings for the escape of the gas in front of the piston until it reaches a certain height, which should conform to the lift of the valve; at this point the piston is prevented from traveling farther by the compression of the gas and the valve is brought to a stop without noise or jar.

The strength of spring for closing the discharge valve must be nicely gauged; if too light the valve will not be brought to its seat until the piston has started on the return stroke, causing a return to the compressor of a portion of the compressed gas, and the striking on the seat with considerable force due to the pressure of the gas above the valve and the removal of the pressure below; while if the spring is too strong the valve is returned with such force as to cause a harsh sound and rapid wearing. This necessitates more frequent renewals than if moved with a spring of the proper tension.

Inasmuch as the weight of the moving part of the valve is a factor in determining the required strength, and the diameter, pitch, temper, gauge of spring and material, determine the strength, it becomes practically impossible to lay down a rule for the selection of the proper spring for each condition. Experience shows that one can better determine what spring to use by the sense of feeling than by any rule or standard.



SKATING RINK BEFORE FLOODING, SHOWING BRINE PIPES

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Piston. Having provided the inlet and outlet valves with proper opening and closing devices and made them capable of retaining the gas passing them, a piston for compressing the gas and discharging it from the compressor must be provided. For the vertical type of single-acting compressor in which both inlet valves are in the upper compressor head, the piston is best made as a ribbed disc with a hub at the center for the piston rod and a periphery of sufficient width to be grooved for the necessary snap rings. Three to five of these rings are generally used.

Fig. 12 illustrates the simplest form of piston of this type, A being the cast head, B the snap rings, and C the piston rod. The surface is faced square with the bore of the hub, and the rod forced in and riveted over, filling the small counterbore provided for this purpose. The cast head and rod having been previously roughed out, is now finished in its assembled condition, the rod made parallel and true to gauge, threaded to fit the crosshead and the grooves turned for the snap rings, which are made slightly larger than the bore of the compressor. A diagonal cut is made through one side and enough of the ring is cut out to allow it to slip into the cylinder without binding. It is then scraped on its sides until it fits accurately the groove in the piston. It is also well to turn a small half-round oil groove in the outer face of the piston between each ring which gathers and retains a portion of the oil used for lubrication, thus increasing the efficiency

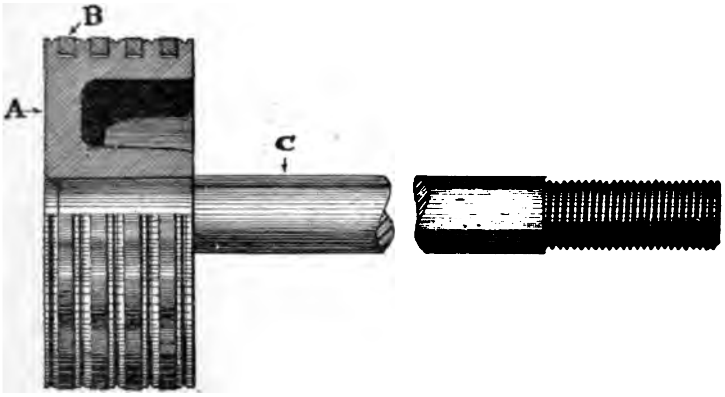

Fig. 12.

of the piston and collecting dust or scale and lessening the liability of cutting of the cylinder due to any of the usual causes.

The piston rod requires special care both in workmanship and material. In order to be effective it must be true from end to end and to be lasting under the variety of conditions which it operates, should be of a good grade of tool steel. The end which is usually made to screw into the crosshead is turned somewhat smaller, usually from $\frac{1}{8}$ to $\frac{1}{4}$ inch in diameter, than the portion passing through the packing or stuffing box principally to allow of returning or truing up the rod when it becomes worn, and also to allow it to pass through the stuffing box. After the rod is screwed into the crosshead it is secured and locked with a nut to prevent turning. The nut also allows the position of the piston to be changed to compensate for wear on the different parts of the machine by simply loosening the lock nut and turning the piston and rod in or out of the crosshead.

The stuffing box of the compressor, shown in Fig. 13, is one of the most difficult parts to keep in proper order. This is owing principally to one of two causes: not being in line with the crosshead guide or bore of the compressor, or the great difference of temperature to which it is subjected owing to the possible changes taking place in the evaporator. However, with the compressor crosshead guides, and stuffing box in perfect alignment, and a constant pressure or temperature on the evaporating side, it is a simple matter with almost any kind of packing on a machine of the vertical single-acting type to keep the stuffing box tight and in perfect condition. If, however, either of the above conditions are changed it becomes practically impossible to accomplish this. We have learned that perfectly constructed and operating valves is one of the essential features of a perfect machine. We also know that the alignment of the machine is equally important.

Erection. Experience in the building, erecting and operation of this class of machinery, shows that more machinery is condemned, more complaints made and difficulties encountered owing to these two points than all others put together. It is all important to the erecting and operating engineer that they be absolutely certain of these two points.

After the A frame, the compressor cylinder and its lower head have been placed in position, a fine hard line should be passed through the cylinder and stuffing box down through the guide and to the crank pin in the shaft, drawn tight and secured in some manner at each end and then callipered at each point. The different parts should be brought into perfect alignment before the rest of the machine is assembled.

Fig. 14 illustrates the methods of obtaining the proper results. Assuming that the brasses are central with the connecting rod and the connecting rod with the crosshead, the line should of course be central with the crank pin and this condition should exist when the pin is at its top and bottom positions. In other words, when the line is placed midway between the collars on the crank pin when at the bottom stroke, and the shaft is revolved one-half revolution to its top stroke, the line should be exactly midway between them; or the top moved until such is the case, and this should be determined as absolutely correct before proceeding further. When this is accomplished we may proceed to the guides and caliper first at the bottom and then the top between the line and each side of the bore, moving the A frame by dressing the bottoms of the feet or packing under, as may be most desirable.

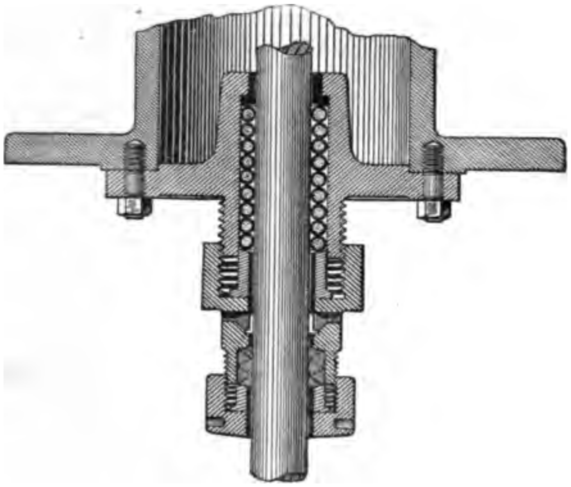

Fig. 13.

We may now examine the top of the compressor and the bottom of the stuffing box to determine whether or not these are central. By shimming or packing under the side of the compressor, the cylinder and the stuffing box may be moved in the required direction; this will be found all that is necessary to correct the small inaccuracies. Of course, should objection be raised to the method, the other remedy is to dress the surfaces.

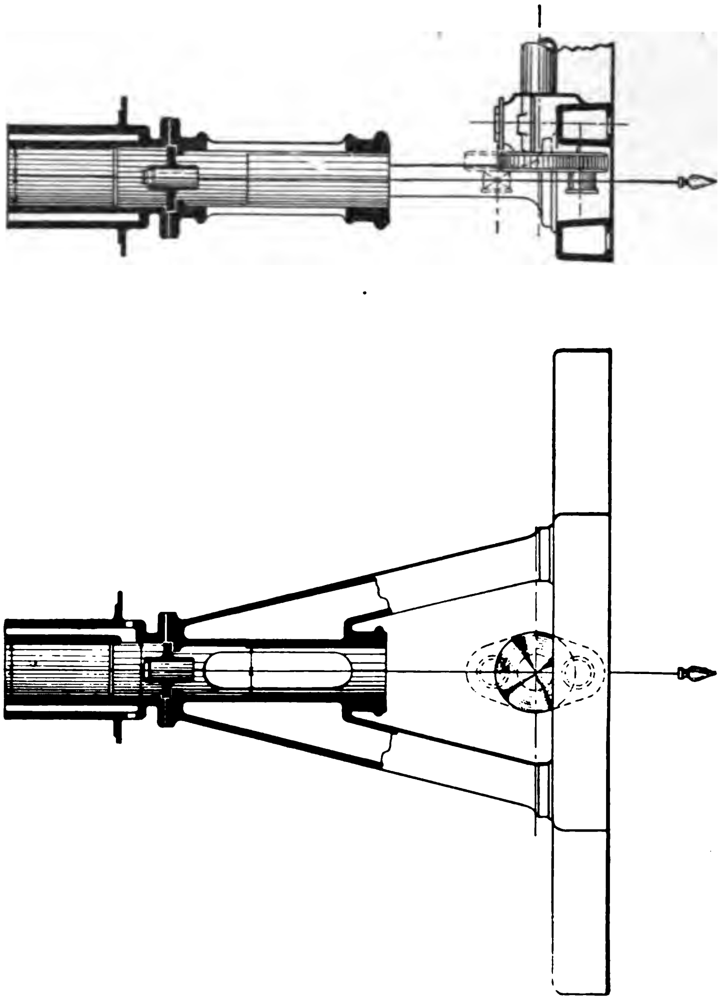

Fig. 14.

Having the different parts in proper alignment, we may now proceed with the assembling of the balance of the machine, feeling assured that with reasonably good workmanship and material, we will have a good running machine. It is well to have a compressor stuffing box of any of the several types in two parts or double packed (see Fig. 13), the inner to be of proper proportion to hold the packing against the loss of ammonia and the outer of only slight depth to retain the lubricating oil within the annular space provided between the two, and through which the rod passes in its travel. The packing may be drawn up or tightened by any one of several common devices, although a parallel device or one which causes the gland to move uniformly by tightening at one point is to be preferred. Although three or four separate bolts are very often used they are undesirable from the fact that they may cause a tipping or "cocking" due to the tightening or loosening of one.

There are innumerable materials for packing, and the "packing man" is encountered every day extolling the "good qualities" of his packing. It is probable that all have good points, but it is doubtful if any one make or kind would meet with the unqualified indorsement of all engineers; also no one packing is best for all conditions or duties. The condition and packing must be suited to one another; this will generally be accomplished by the good engineer.

In the construction of the stuffing box it is well to bush the bottom of the glands with Babbitt metal, because the rod is often pressed to one side crowding it against the side of the gland or bottom of box, causing the same to cut. When it becomes necessary to turn down the rod, a new bushing is all that is necessary to reduce the openings correspondingly.

Water Jacket. In the vertical single-acting type of compressor, it is usual to provide a water jacket, which may be cast in combination with the compressor cylinder or made of some sheet metal secured to an angle, which is bolted to a flange cast on the cylinder. It is usual to have this water jacket start at about the middle of the compressor (or a little below as shown in Fig. 15) and extend enough above to cover the compressor heads, valves and bonnets with water; the principal object of which is to keep these parts at a normal temperature and thereby improve the operation as well as protect the joints against the excessive heat which would be generated by the continued compression. It is also an advantage in the operation of the plant, since by reducing the temperature in the compressor and adjacent parts, the compressor is filled with gas of a greater density. It is also true that the heat extracted or taken up by the water at this point is a certain portion of the work performed in the condenser and therefore not a waste.

In the operation of the plant it is well to have plenty of water flow through the jackets, as the cooler the compressors are kept the better, but in plants in which water is scarce the quantity may be reduced correspondingly until the overflow is upwards of 100° Fahr. In extreme cases of shortage of water the overflow water from the ammonia condenser is sometimes used on the water jackets, that is, the entire amount of available water is delivered to the condenser, and a supply from the catch pan (if it be an atmospheric type) is taken for the water jackets, in which case a greater quantity may be used but at a higher temperature. It is customary to admit the water through the flange forming the bottom of the water jacket and overflow near the top into a stand pipe which is connected at its lower end through the flange to a system of pipes to take it away. To prevent condensation on the outer surface of the jacket, and to present a more pleasing appearance, it is frequently lagged with hardwood strips and bound with finished brass or nickel-plated bands. It is also well to have a washout connection from each jacket.

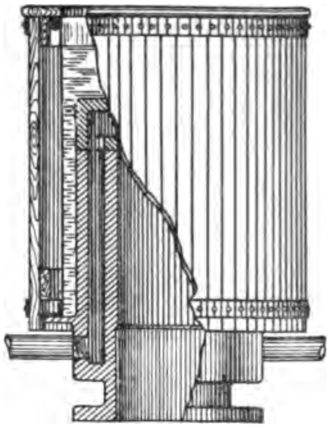

Fig. 15.

Lubrication. The vertical type of compressor requires the least amount of lubrication from the fact that all moving parts are in equilibrium. The slight amount of oil used is merely to keep the surfaces from becoming entirely dry. Excessive lubrication is an objection, owing to the insulating effect upon the surfaces of the condensing and evaporating system. Therefore it is well to feed to the compressors as little as is consistent with the operation of the machinery. A proper separating device should be located in the discharge pipe from the compressor to the condenser. To properly admit, or feed the lubricant to the compressors, sight feed lubricators should be provided, by which the amount may be determined and regulated. These may be of the reservoir type, or better still the droppers, fed from a large reservoir through a pipe and which may be filled by a hand pump when necessary (see Fig. 16). Owing to the action of ammonia on animal or vegetable oils, other than these must be used as a lubricant for the compressor. The principal oil for this purpose (and when obtained pure, a very good one) is the West Virginia Natural Lubricating Oil or Mount Farm, which is a dark-colored oil not affected by the action of the ammonia or the low temperature of the evaporator. Of late years the oil refining companies have put on the market a light-colored oil which appears to give good results for the purpose. Care should be used, however, ir

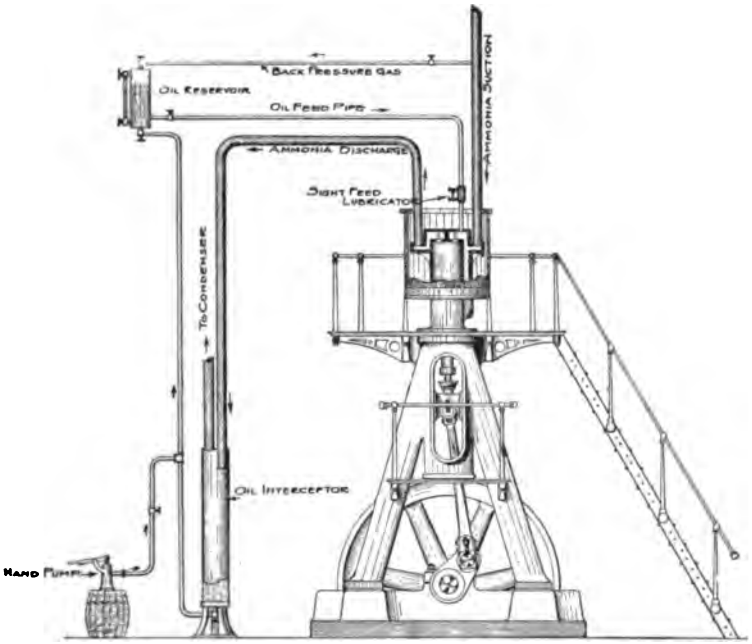

Fig. 16

the ammonia and oil should not be used unless it is of the proper grade, as serious results follow the use of inferior oils. The usual result is the gumming of the compressors and valves or the saponifying under the action of the ammonia through the system.

LOSSES.

Having described the compressor and its parts, let us take up the losses due to the improper working or assembling of the parts of the machine, before proceeding with the description of the rest of the plant. As has been stated in a general way, the economy of the compressor lies in its filling at the nearest possible point to the evaporating pressure and then compressing and discharging at the lowest possible pressure, as much of the entire contents of the cylinder as possible. If the compressor piston does not travel close to the upper end of a single-acting machine, or [illegible] has excessive clearance, the compressed gas [illegible] cylinder re-expands on the downward stroke of [illegible] gas from the evaporator will not be taken into the [illegible] the pressure falls to or slightly below, this point. [illegible] this fault is equal to the quantity of gas thus [illegible] the compressor plus the friction of the [illegible] the portion of the gas thus expanding. [illegible] discharge of the gas and there is a leaky [illegible] the escape and re-expansion into [illegible] the intake of the gas at the beginning [illegible] continues to affect the amount of [illegible] the entire stroke and the capacity of the machine [illegible] reduced. If the inlet valve is leaky [illegible] becomes lodged on its seat, as the [illegible] the portion of the gas which may escape [illegible] compression is forced back to the evaporator [illegible] loss is the result. A piston which does not [illegible] piston rings, or a compressor which has [illegible] the point of allowing the escape of gas [illegible] has the same effect as the ill [illegible] The loss due to leaky or defective [illegible] stuffing boxes, are not included under this head as these [illegible] generally affect the loss of the material than the efficiency of the compressor.

To present graphically the losses due to the above causes refer to Fig. 17, which illustrates the usual indicator card taken from an ammonia compressor and the effect upon it of the various losses.

The different mechanisms employed in actuating the compressor piston, such as crossheads, connecting rods, pins, crank shaft and bearings, are of usual engine practice. A detailed description will not be given here. It should be stated that owing to the steady load and the continuous service usually demanded of a refrigerating plant, and the occasional high pressures encountered in the extremely warm weather, the construction should be of the best and the wearing surfaces ample to withstand the hard service. The many types of machines and the methods of connecting the power or engine are of almost endless variety. It is sufficient to state that the general classes of engines are of vertical or horizontal single-cylinder, tandem or cross compound, condensing or non-condensing.

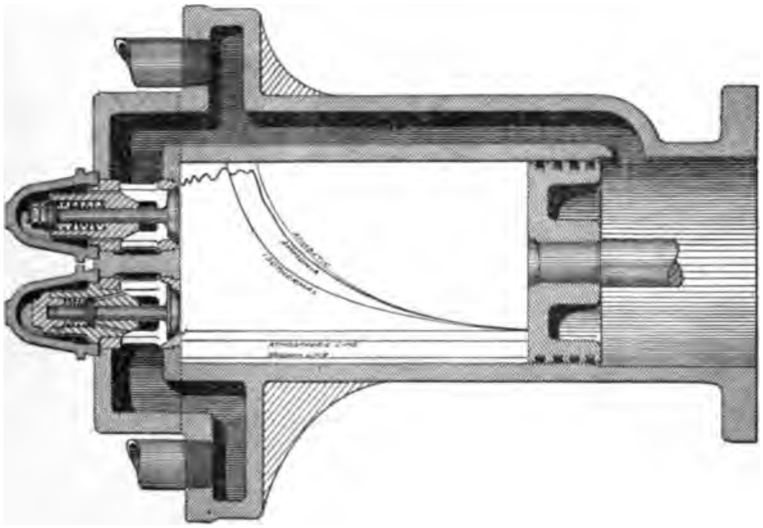

Fig. 17.

The horizontal double-acting compressor embodies all the general principles of the vertical single-acting, but having certain modifications to meet the mechanical differences presented. They may be divided into two classes, those having water jackets and those without. The former usually has a water jacket extending about the body of the cylinder but not over the heads and valves as in the single-acting compressor. Compressors not provided with water jackets are internally cooled either by evaporation of a portion of the ammonia, or the injection of ammonia or cooled oil. The claims of both these last named systems are that instead of the curve of compression being close to the adiabatic it is brought nearly to the isothermal, and that a corresponding economy in horse power required to operate the machine is effected. Owing to the fact that this type of compressor takes in and discharges gas in both directions, it is necessary that inlet and outlet valves be provided in both heads, and that the inner head be provided with a stuffing box also for the piston rod. This renders it impossible to place the valves horizontally, and as a result, the valves are placed at an angle of from 30° to 45° from the center of the compressor, and the inner face of the heads and pistons made on an angle to conform to this; or sometimes the valves are spherical, to avoid clearance which would result from the difference in their surfaces. The weight of the piston and rod being towards the bottom of the cylinder, it is necessary that the piston be of considerable length to provide a greater surface for wear.

The stuffing box must have sufficient depth to successfully retain the gas during the period of compression, and lubrication should take place while the rod is passing through the packing. This is accomplished with a small oil pump actuated from the moving parts of the machine or a pressure system regulated by the pressure in the oil separator In each case oil passes between the double packing thereby causing the piston rod to pass through a chamber of oil.

The compressor valves and their cages are of the description already given and the same care and caution regarding the construction and operation apply to the valves of this type of machine, but the fact that they are horizontal or nearly so makes their construction somewhat different and necessitates springs of different strength. The valves, cages and springs are covered by a dome bolted in place and its union with the compressor head is provided with a gasket to prevent loss of ammonia. It is well to have at hand duplicates of the compressor valves and springs, as an entire valve can be changed in less time than it takes to repair one out of order. As the machine during this time must remain out of service, it is customary for the builders to supply these in duplicate. After a valve is taken out and replaced, it is well to have the faulty one put in perfect condition, the strength of springs tested, the seat and stem corrected, oiled and put away in readiness to be used again when the occasion requires.

This type of compressor is usually driven from a rotating shaft with the ordinary connecting rod by a crank and crosshead. Adjustment for wear is provided in the piston rod at the crosshead, and the position of the piston may be adjusted between the heads of the compressor by screwing in or out of the crosshead. The clearance of the compressor piston is important; it should be as small as possible and yet not allow the piston to strike the heads. This naturally cannot be reduced to the small degree possible in the single-acting type, from the fact that it must be adjusted to both heads, and the expansion and contraction of the piston and connecting rod and wear is of such an amount as to render a close adjustment impossible. Fig. 18 is a sectional cut of a modern type of double-acting compressor with water jacket; Fig. 4 a section of one not using the water jacket and Fig. 10 an elevation of either in combination with its motive power, a Corliss engine, shaft, fly wheel, etc.

The stuffing-box oil pump is shown in Fig. 16 which illustrates the general method of lubricating the compressor piston, the oil being pumped through the box and returning to a catch pan or basin provided in the bed plate.

THE AMMONIA CONDENSER.

The Ammonia Condenser, or liquefier, as briefly stated in the description of the system, is that portion of the plant in which the gas from the evaporator, having been compressed to a certain point, is cooled by water and thereby deprived of the heat which it took up during evaporation; consequently it is reduced to its initial state, that is, liquid anhydrous ammonia. Let us consider the general principles governing the action before describing the types. On account of the duty having been performed, the ammonia as it leaves the evaporator is a gas of low temperature,

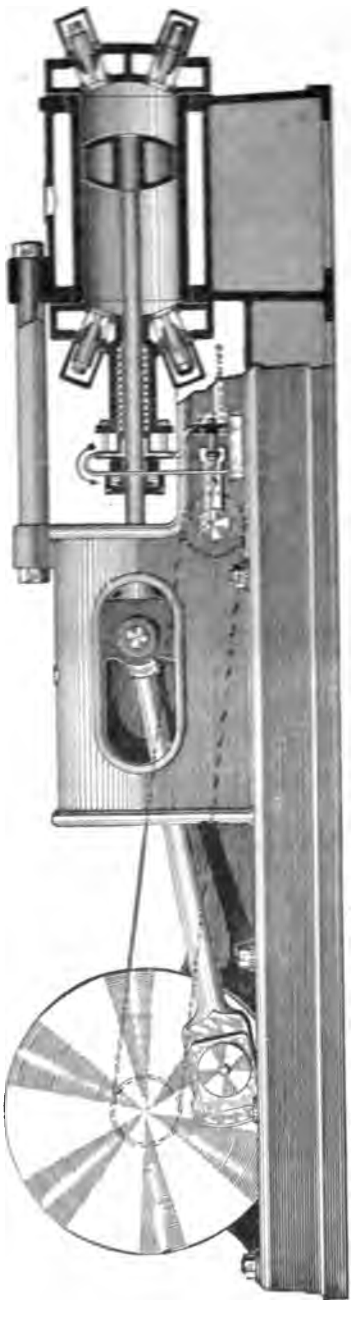

Fig. 18.

usually 5° to 10° below that of the brine, or other body upon which it has been doing duty, yet it is laden with a certain amount of heat, although at a temperature not ordinarily expressed by that term. It is a well-known fact that we cannot obtain a refrigerating agent which can absorb heat from a body colder than itself, and it is therefore necessary to bring the temperature of the ammonia gas to a point at which the flow of heat from the one to the other will take place. This is done by withdrawing part of the heat in the ammonia in the following manner: The cold gas is compressed until its pressure reaches such a point that at ordinary temperatures it will condense to liquid form. As it leaves the compressor it is very hot because of the fact that it still contains nearly all of the heat it had when it left the evaporator in only a small portion of the space occupied before. Thus when it reaches the condenser it is much warmer than the cooling water and will readily give up its heat to the cold water—so much that its latent heat is absorbed by the water and it condenses into anhydrous ammonia.

The temperature of water if pumped from surface streams will average about 60° F and since we cannot expect to get the ammonia any colder than this it must be compressed until the boiling point corresponding to the pressure obtained is at about 75° F.

In the table, page 78, we find that this temperature corresponds to a pressure of 144.25 pounds per square inch (absolute) or 126.55 pounds per square inch (gauge).

Thus if the gas is compressed until the gauge reads 126.55 and then passed into a condenser where the temperature of the water is less than 75° F the water will absorb the latent heat and we have accomplished our object which was to remove some of the heat contained in the ammonia. In this condition it is drained from the condenser into the ammonia receiver to again repeat the cycle of operation.

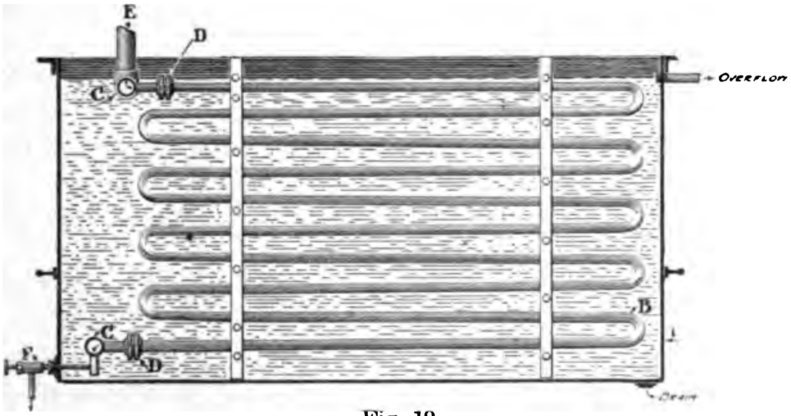

Fig. 19.

The forms of condenser may be divided into three classes; the submerged, atmospheric, and double-pipe. Of each of these classes a number of different types and constructions are in use. To illustrate the general principles, however. it is only necessary to present one of each type.

The Submerged Condenser consists of a round or rectangular tank with a series of spiral or flat coils within joined to headers at top and bottom with proper ammonia unions. In Fig. 19 is shown a sectional elevation of a popular type of submerged condenser. A wrought iron or steel tank A is formed by plates from $\frac{3}{16}$ to $\frac{5}{16}$ inch thick, of the necessary dimensions to contain the coils, and sufficiently braced around the top and sides to prevent bulging when filled with water. A series of welded zigzag pipe coils B are placed in the tank and joined to headers C with ammonia unions D. The ammonia gas enters the top header through the pipe E and an outlet for the liquefied ammonia is provided at F with a proper stop valve. Water is discharged or admitted to the tank at or near the bottom and overflows at outlet

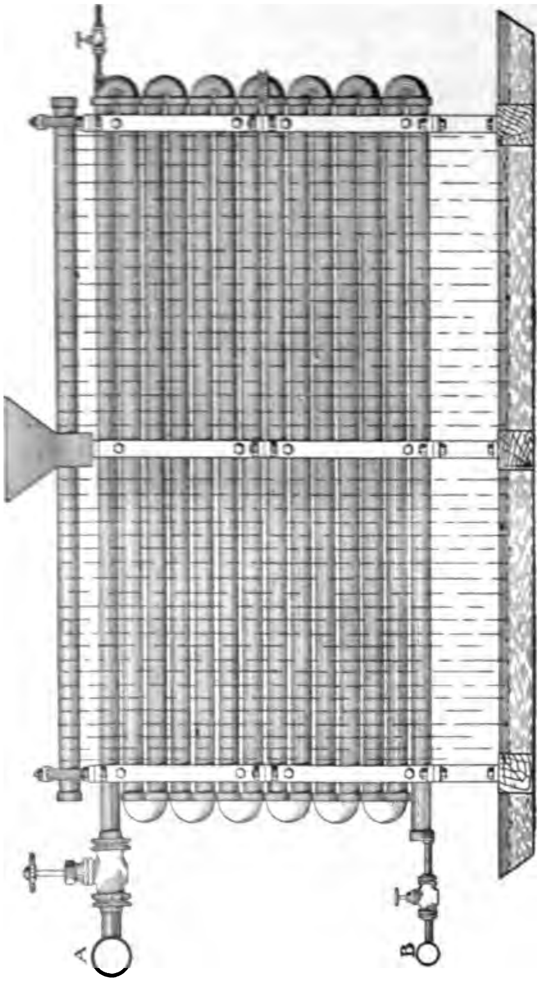

Fig. 20.

M. It will be seen that in this type of condenser a complete reverse flow of the current is effected, the gas entering at the top and the liquid leaving at the bottom, while the water enters at the bottom and leaves at the top, thereby bringing the coldest water in contact with the coolest gas and the warmer water in contact with the incoming or discharged gas from the compressor, thereby presenting the ideal condition for properly condensing ammonia.

Owing to the necessarily large spaces between the coils and the distance between the bent pipes, the portion of water coming in contact with the surface of the pipes must be small compared with the total amount passing through; it is, therefore, uneconomical as regards amount of water used. With water containing a large amount of floating impurities the deposit on the coils is considerable, and not easily removed owing to the limited space between the coils, and furthermore, the dimensions of the tank necessary to contain the requisite amount of pipe for a plant of considerable size is so great and its weight when equipped with coils and filled with water requires such a strong support that its use is now limited to certain requirements and localities.

A better shape for a condenser of this type is one of considerable height or depth, rather than low and broad. This is owing to the fact that the greater the length of travel of the water and gas in opposite directions, the greater the economy. The number of coils used should be such that the combined internal area of the pipes equals, or exceeds the area of the discharge pipe from the compressor. The circular submerged condenser is similar to the above described except that the tank is circular and the coils bent spirally.

The Atmospheric type of condenser most generally used is made of straight lengths of 2-inch extra-strong, or special pipe, usually 20 feet long, screwed, or screwed and soldered into steel return bends about 3½-inch centers and usually from eighteen to twenty-four pipes high. The coil is supported on cast or wrought iron stands and placed within a catch pan, or on a water-tight floor, having a proper waste water outlet and supplied with one of the several means of supplying the cooling water over their surfaces. Stop valves, manifolds and unions connect with the discharge of the compressor and the liquid ammonia supply to the receiver.

In the manner of making the connections to this type of condenser and the taking away of the liquefied ammonia as well as in the devices for supplying the cooling water, a great variety exists; Fig. 20 represents a side elevation of an ammonia condenser with

[illegible]

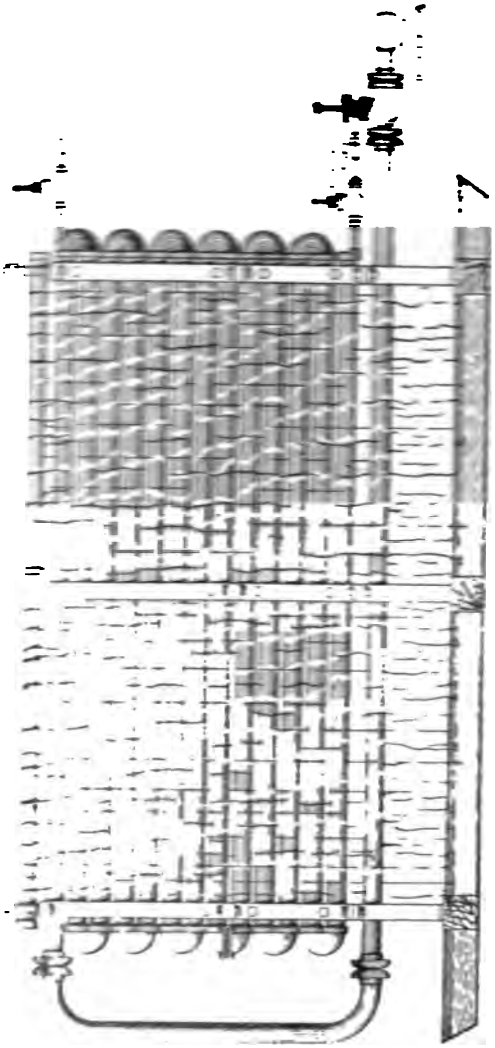

Fig. 21.

catch-pan or water-tight floor where it accumulates and is taken away by any of the usual means. It will be noticed that the flow of the water and gas with this type of condenser is in the same direction, the coldest water coming in contact with the warmest

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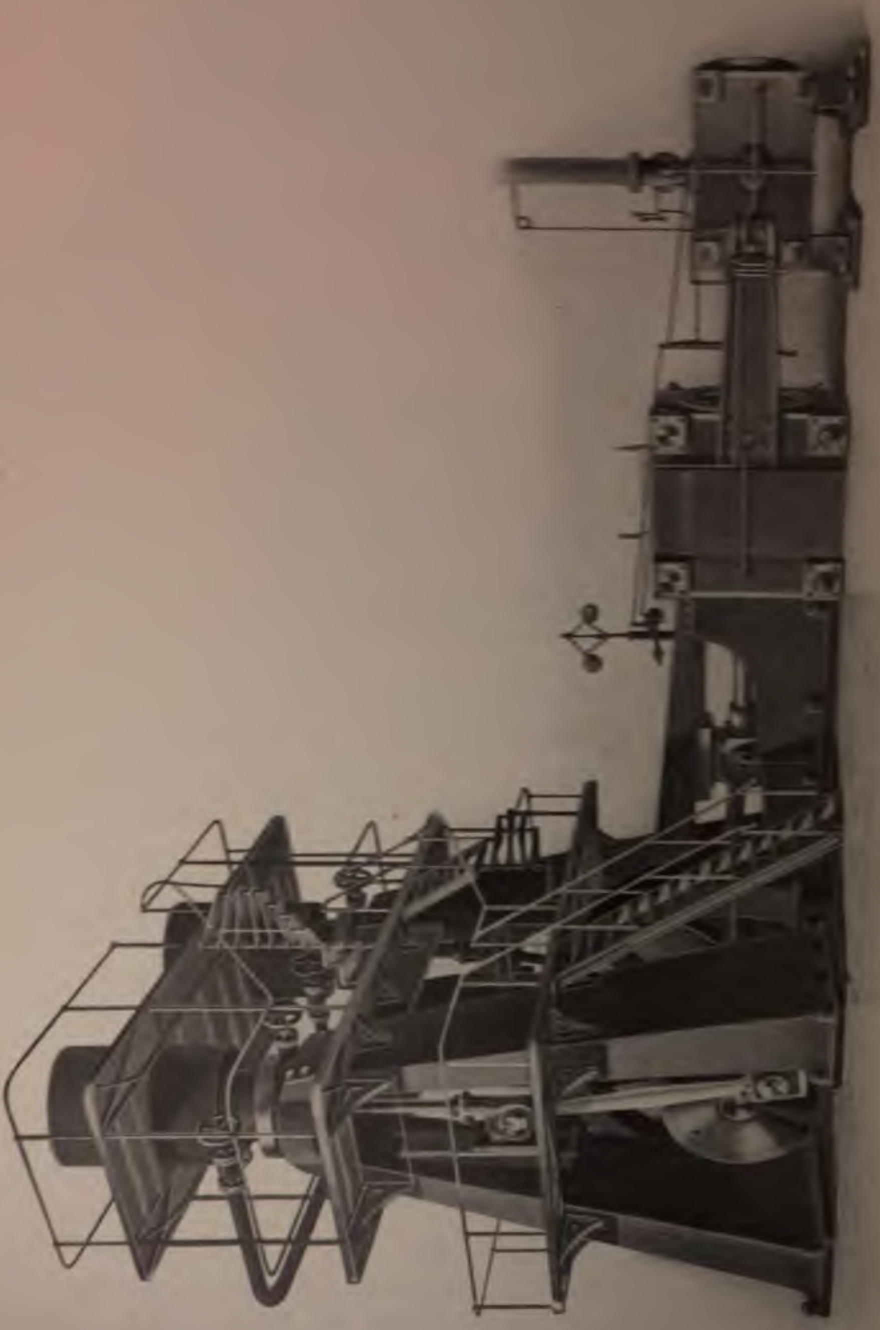


ammonia. The temperature governing or determining the point of condensation will be that of leaving the condenser, or at the bottom pipe in which the liquid ammonia is withdrawn. Owing to this arrangement it is not favorable to a low condensing pressure or economy in the water used. Fig. 21 represents a type in which an attempt is made to eliminate this undesirable feature, and in which it is expected to use the waste from the condenser proper, in taking out the greater part of the sensible heat from the gas leaving the compressor.

The construction of this condenser is identical with that shown in the preceding figure except that its uppermost pipe is continued down and under the pipes forming the condenser proper; it passes backward and forward in order that a large proportion of the heat may be removed by the water from the condenser proper, before the ammonia enters the condenser. A supplemental header is sometimes introduced in connection with this pipe for removing any condensation taking place in it.

A third type of this condenser is shown in Fig. 22. In this type a reverse flow of the gas and water takes place. The gas enters the condenser through a manifold or header A at the bottom and continues its flow upward through the pipes to the top; at several points drain pipes are provided for taking off the condensation into the header B. The condensing water flows downward over the pipes. The form is the most nearly perfect condenser of the class.

The atmospheric type is a favorite, and possesses many features that make it preferred to the submerged. Its weight is a minimum, being only that of pipe and supports and a small amount of water. The sections or banks may be placed a favorable distance apart to facilitate cleaning and repairs, while the atmospheric effect in evaporating a portion of the condensing water during its flow over the condenser, thereby obtaining the advantage of its latent heat as well as the natural raise in its temperature, adds materially to its efficiency.

The various devices for distributing the water over the condenser are numerous:

Fig. 23 represents the simplest and most easily obtained; a simple trough with perforations at the bottom for allowing the water to drip through to the condenser

Fig. 24 is a modification of the one shown in Fig. 23. This is intended to prevent the clogging of the perforations, by allowing the water to flow into the space at one side of the partition, and then through a series of perforations into the second, and thence through a second set of perforations in the bottom to the pipes in the condenser.

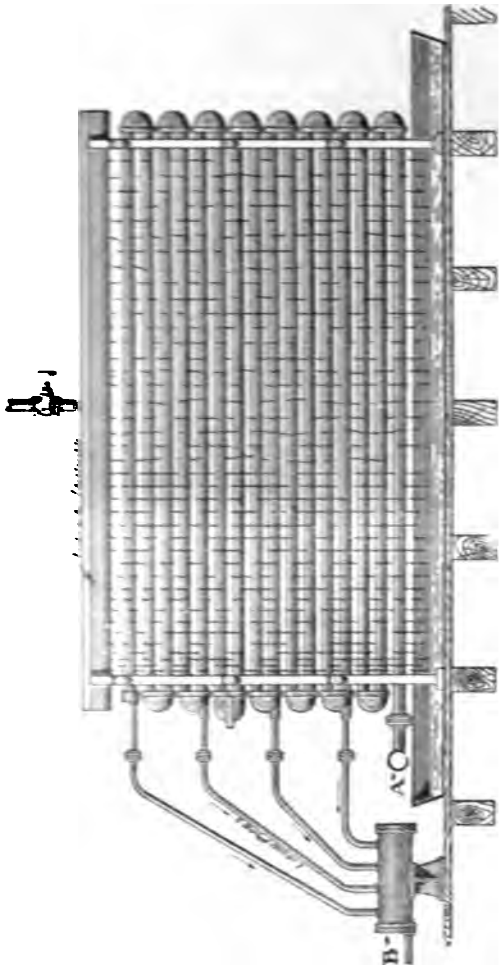


Fig. 22

Fig. 25 is a type of trough, or water distributor, designed to overcome the objections to a perforated form of trough, and the consequent difficulties due to the clogging or filling of the perforations. As will be readily understood from the illustration, this is also made of galvanized sheet metal with one side enough higher than the other to cause the water to overflow through the V-shaped notches or openings along the top of the straight or vertical side of the trough, and down and off the serrated bottom edge to the pipes.

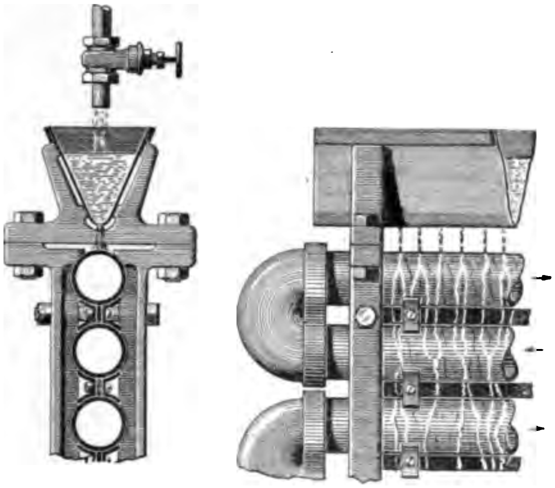


Fig. 23.

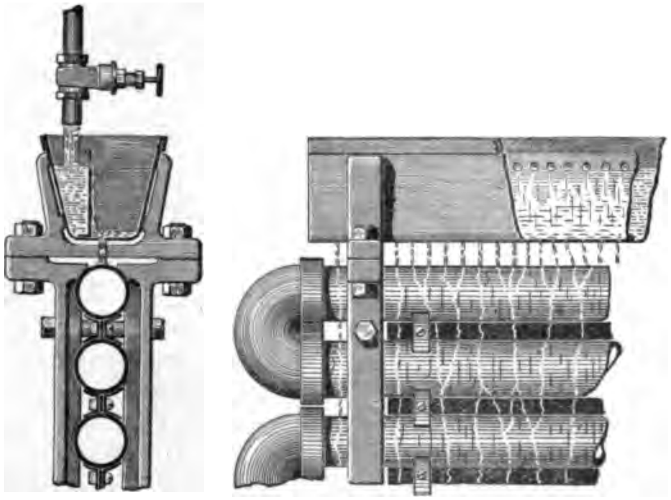


Fig. 24.

The object of the serrated edges, as will be apparent, is the more even distribution of the water, owing to the fact that while it would be practically impossible to obtain a uniform flow of water over a straight and even edge of a trough, particularly if the amount is limited, it is an easy matter to regulate the flow through the V-shaped openings.

Fig. 26 is termed the "slotted water pipe." It is a pipe slotted between its two ends, from which the water overflows to the series of pipes below. It is good practice to lead the water supply to a cast-iron box at the center of the condenser, into the sides of which is screwed a piece of pipe (usually 2-inch) reaching the ends of the condenser, and having its outer ends capped, which may be removed while a scraper is passed through the slot from the center towards the ends while the water is still flowing, thereby carrying off any deposit within the pipe. This forms very durable construction, and not liable, as with the galvanized drip trough, to disarrangement or bending out of shape due to various causes. It is impossible, however, to obtain the uniform flow of water over the condenser with this, as with the serrated or perforated troughs, particularly if the supply is limited, from the fact as stated in describing the overflow trough, viz: the impossibility of obtaining a sufficiently thin stream of that length.

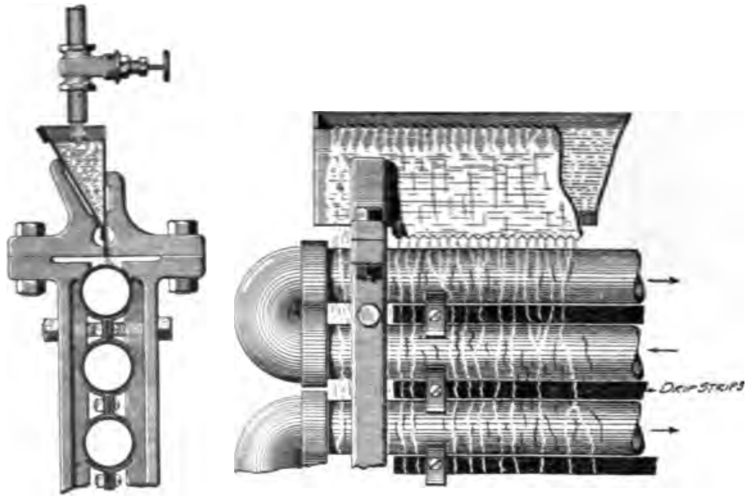


Fig. 25.

The Double-Pipe Condenser is a modern adaptation of an old idea, given up owing to its complex construction and the imperfect facilities available for its manufacture. Also, like the Brine Cooler, it has come into use with great rapidity, and has brought forth many novel ideas of principle and construction. It combines the good features of the atmospheric as well as the submerged; as in the former the weight is small and it is accessible for repairs. It has the downward flow of the ammonia and upward flow of the water, effecting a complete counter flow of the two. minimizing the amount of water required and taking up the heat of condensation with the least possible difference between the ammonia and water.

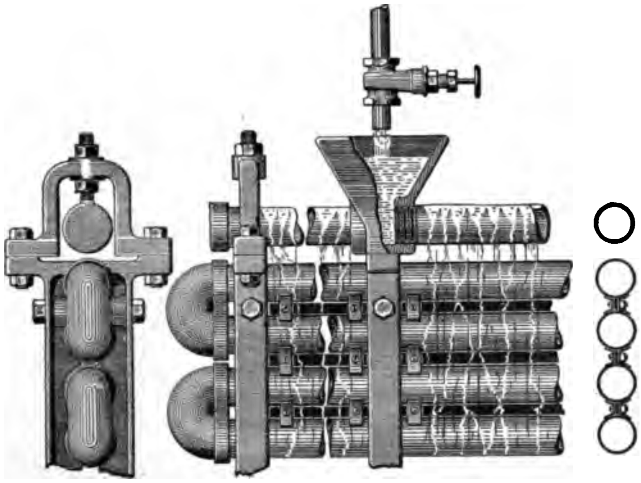


Fig. 26.

The two general forms of construction are a combination of a 1¼-inch pipe within a 2-inch pipe, or a 2-inch pipe within a 3-inch. The water passes upward through the inner pipe, while the gas is discharged downward through the annular space; or, the position of the two may be reversed, the ammonia being within the inside pipe while the water travels upward through the annular space. They are also constructed in series, in which the gas enters a number of pipes of a section at one time, flowing through these to the opposite end to a header or manifold, at which point the number of pipes is reduced, and so on to the

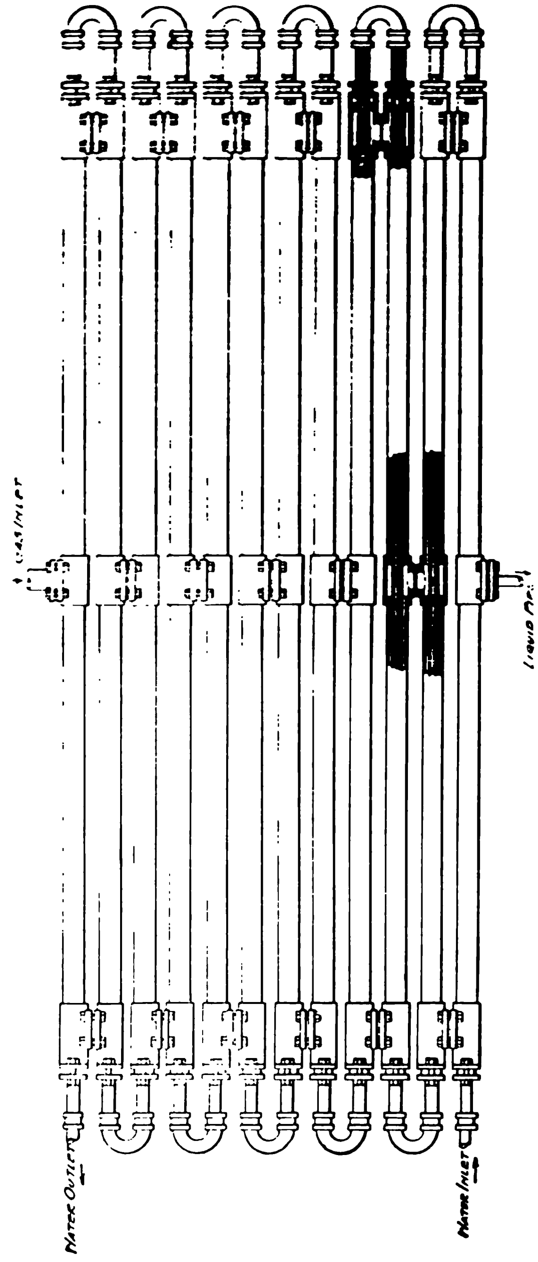


Fig. 27

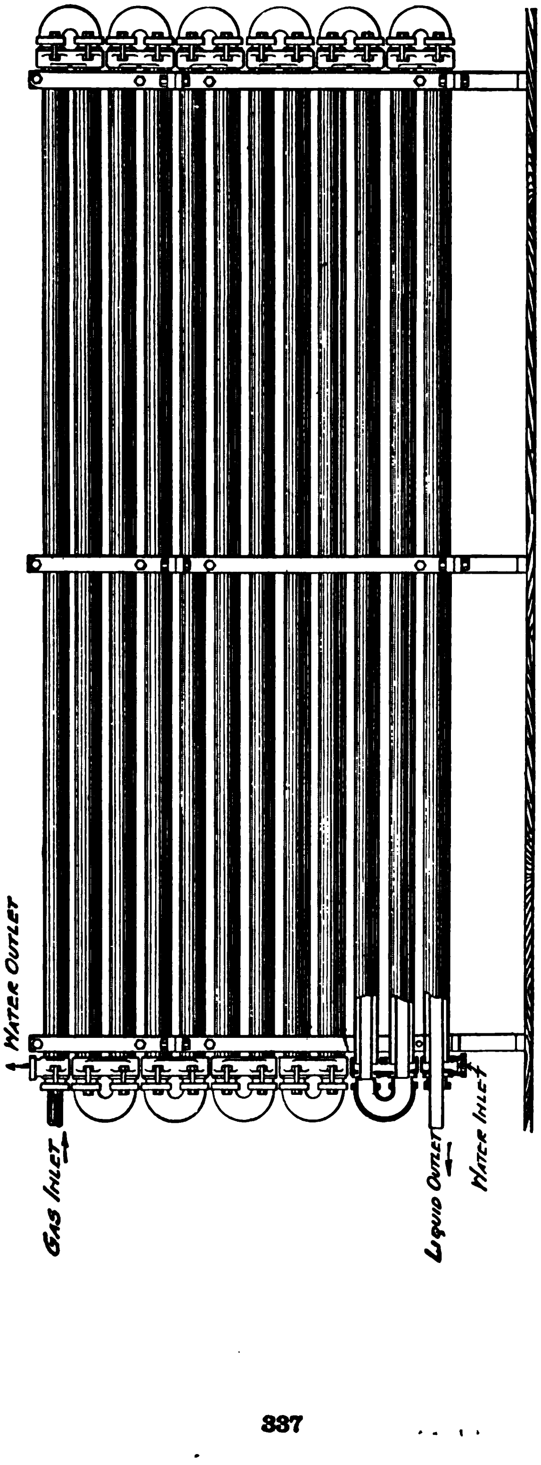


Fig. 28.

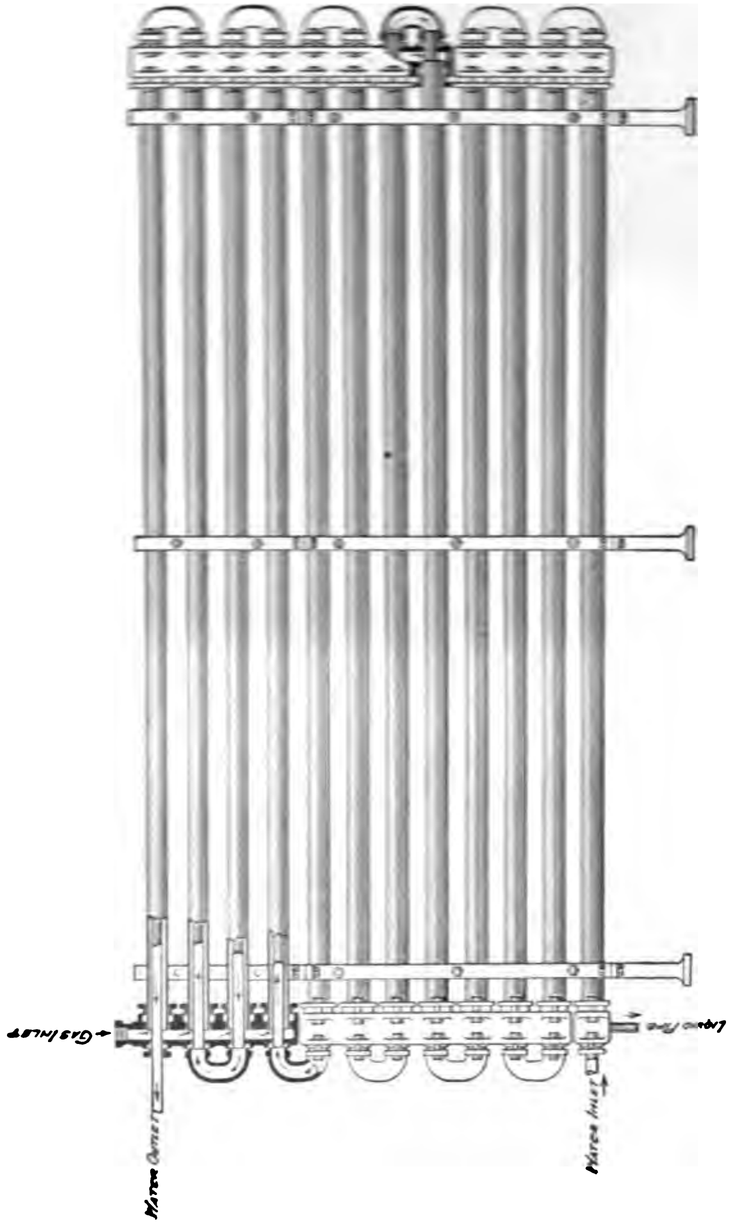


Fig. 20.

vith a constantly reduced area. The theory of this con-
ı being that the volume of the gas is constantly reduced
ıeing condensed.

s. 27, 28 and 29 illustrate a general range of the various
use.

is usual in the construction of this type to make each
ɔr bank twelve pipes high
.t 17½ feet long; they are
ɔminally at ten tons refrig-
capacity each, although for
units the construction is
vary from 10 to 14 pipes

il Separator or Interceptor
·ice, or form of trap, placed
ıe of the discharge between
ıressor and the condenser to
the oil from the ammonia
is to prevent the pipe sur-
he condensing and evapor-
tem from becoming covered
which acts as an insulator
'ents rapid transmission of
ıugh the walls.

ınstruction admits of quite
of principle, from the plain
al shell with an inlet at one
end and an outlet at the
the almost endless variety
plates, spiral conductors
ɔrse-current devices. The
similar to that of the steam
.st separator, and generally speaking, that which would
.ve in one service would be so in the other. Figs. 30, 31
ustrate three of the most common types in use; from these
ınt will understand the general principles.

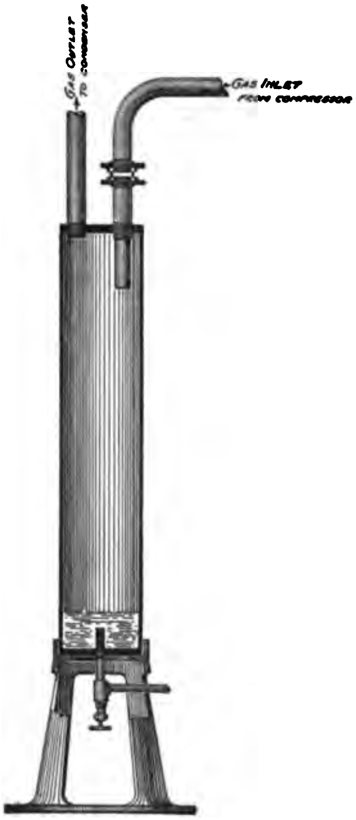


Fig. 30.

AMMONIA RECEIVER.

Ammonia Receiver or Storage Tank is a cylindrical shell with heads bolted or screwed on, or welded in each end, and provided with the necessary openings for the inlet and outlet of the ammonia, purge-valve and gauge fittings. They may be vertical or horizontal; the former type is generally used on account of the saving of floor space, while the horizontal is necessary when the condenser is located so low as to make the flow of the liquid ammonia into the vertical type impossible. A convenient location for the receiver in a plant in which the condenser is located above the machine room, is against the wall, or at one side of the room on a bracket or stand at one side of the oil interceptor, the sizes of the two being generally the same. They are then more readily under the control of the engineer than if at some out of the way place.

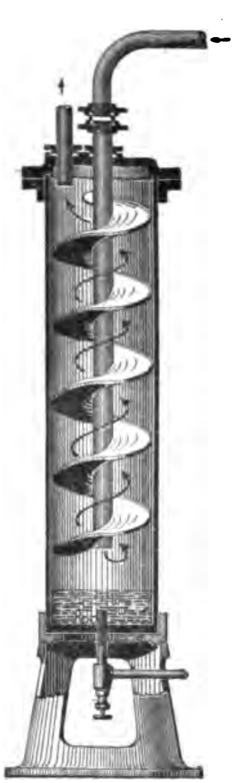


Fig. 31.

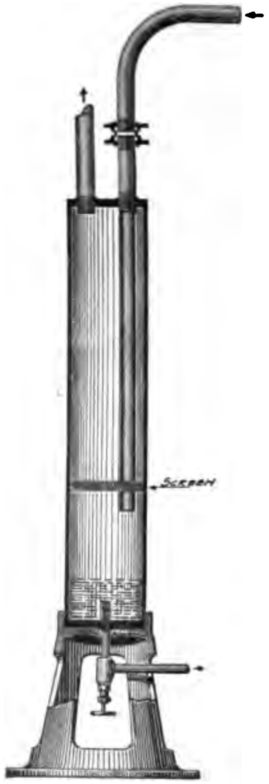


Fig. 32.

Fig. 33 illustrates a receiver of the vertical type with the usual valves and connections for the proper equipment. The liquid ammonia enters at the top and is fed to the evaporator from the side near the bottom, the space below this opening being provided for the accumulation of scale, dirt, or oil, and means are provided for drawing off through the purge valve in the bottom.

PIPES, VALVES AND FITTINGS.

Pipes. Extra strong, or extra heavy pipe (so called) is the generally accepted pipe for connecting the various parts of the refrigerating system. Wrought-iron pipe is generally preferred to steel. Frequently, however, and particularly for the evaporating or low-pressure side of the system, a special weight or grade of pipe is used, also standard or common pipe is sometimes employed for this purpose. Without knowing the particular conditions under which this is to be used, or the relative value of the material, or manner in which the pipe is made, it is always better to use and insist on having the standard extra-strong grade. The threads should be carefully cut, with a good sharp die, making sure that the top and bottom of the threads are sharp and true. With this precaution, and an equally good thread in the fitting, it is not difficult to form a good and lasting joint. Particular care should also be taken that the pipe screws into the fitting the proper distance, and forms a contact the entire length, rather than to screw up against a shoulder without a perfect fit in the thread. This latter often causes leaky joints sometime after the plant has been operated; the temporary joint formed by screwing in too deep against the shoulder or ill-fitting threads very often passes the test and is used for some time after until the combined effect of heat and cold, and action of the ammonia cause it to break out. It is a safe rule that no amount of solder or other doctoring that is not backed up by a good fitting thread to support it can make an ammonia joint. This is particularly true of the discharge or compression side of the plant.

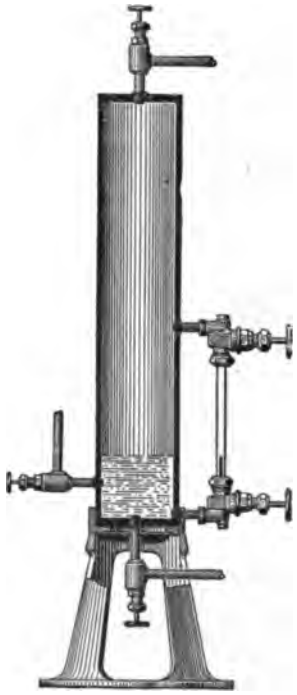


Fig. 33.

The manner of making these joints, may be divided into those having a compressible gasket between the thread on the pipe and the fitting into which it screws and the screwed joint formed by a threaded pipe screwed into a tapped flange or fitting. The latter may be divided into those having a soldered joint, or one in which the union is formed by the ferrule only, with some of the usual cements to assist in making a tight joint.

The two most prominent types of gasket fittings are shown in Figs. 34 and 35. The former is known in this country as the Boyle Union, and is extensively used. As will be observed, the drawing together of the two glands by the bolts compresses the gaskets (usually rubber) against the threaded side of the pipe, the bottom and sides of the recess in the flanges, and the edges of the ferrule between the two gaskets.

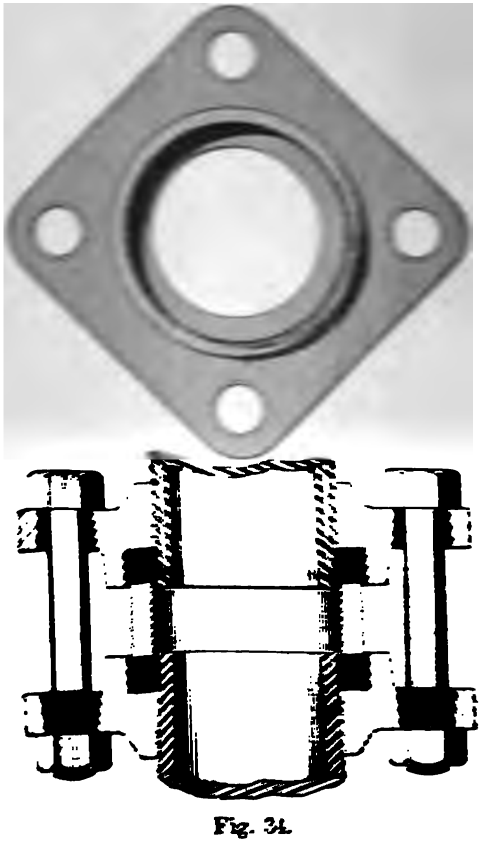


Fig. 34.

Fig. 35 represents a union or joint quite frequently used, although not as commonly as the former. In this the pipe is threaded and screwed into the body of the fitting, but not to form an ammonia-tight joint. Leakage is prevented by a packing ring compressed by the gland against the pipe thread and the walls of the recess.

In Fig. 36 (a type of ammonia coupling), the contact between the pipe and fitting is made to withstand the leakage of the gas without the aid of packing or other material other than solder or some of the usual cements; the two flanges are bolted together with a tongue and grooved joint having a soft metal gasket. This makes a permanent and durable fitting.

Other fittings of the class, as ells, tees, and return bends, are usually provided with one of the above methods of connecting with the system, and the different types described may be obtained of the builders of refrigerating machines.

Valves for the ammonia system of a refrigerating plant are of special make and construction, being of steel or semi-steel, with a soft metal seat which may be renewed when worn, and metal gaskets between the bonnets and flanges.

The usual types are Globe, Angle and Gate, subdivided into screwed and flanged. Fig. 37 is a generally adopted type of the flanged globe ammonia valve, while Fig. 38 represents the angle valve of the same construction. This seems to represent the best elements of a durable and efficient valve.

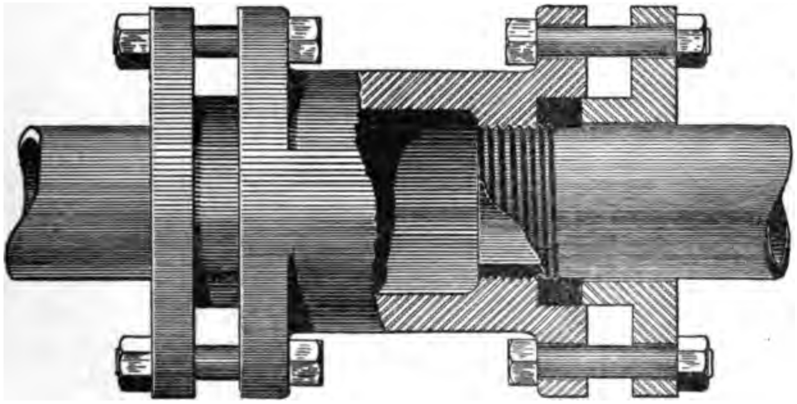


Fig. 35.

For a valve or cock requiring a fine adjustment as is frequently the case in direct-expansion systems, particularly where the length of the evaporating coil or system is short, a V-shaped opening is desirable. Fig. 39 represents a cock for this purpose which will be found to be effective and meet the most exacting requirements.

Pressure Gauges. Two gauges are necessary for an ammonia plant of a single system; one to indicate the discharge or condensing pressure, and one for the evaporator or return gas pressure to the compressors.

Owing to the action of ammonia on brass and copper the gauges for this purpose differ from the ordinary pressure gauge in that it is made with a tube and connections of steel instead of brass, and this construction is the general choice of gauge makers; in other respects the construction is similar. For machines of small capacity instruments with 6-inch dials are common, while for larger plants 8-inch is the generally adopted size. The graduation for the high pressure gauge is usually to 300 pounds pressure, and if a compound gauge is used, it is made to read to a vacuum also. This latter is only needed on certain occasions and frequently omitted from the high-pressure gauge. Owing to the necessity of removing the contents of the system at certain times and usually through the evaporating side of the plant, the gauge for this portion of the system is graduated to read from a vacuum to 120 pounds pressure.

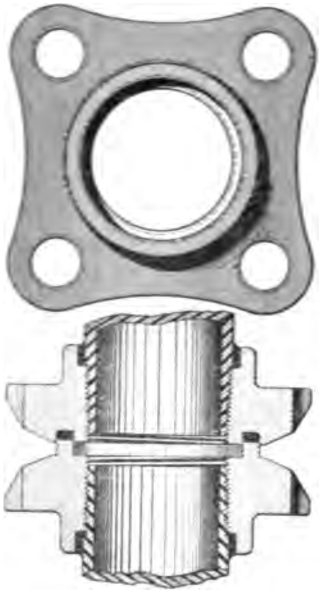


Fig. 36.

In connecting the gauges to the system, it is customary to locate the opening in the discharge and return gas lines near the machine within the engine room, placing a stop valve at some convenient point and carrying a line of $\frac{1}{4}$ or $\frac{1}{2}$ inch extra strong pipe to the gauges, making the joints with the usual ammonia unions. On account of the possibility of leakage of ammonia gas from the gauge tube, it is often considered advisable to fill the gauge pipe with oil (of the kind used for lubricating the ammonia compressor) for a short distance above the gauges, upon which the pressure of the gas will act, causing the gauge to move properly but without allowing the ammonia gas to enter the gauge. This is an application of the same principle as the steam syphon or bent-pipe arrangement in use with steam gauges, for the purpose of keeping the heat and action of steam from the gauge mechanism by the retaining of water in the gauge connection.

Other gauges used about the refrigerating plant are of the ordinary pressure or vacuum types and do not need a special description, as their construction and manner of applying to the different parts of the system are well known to the engineer. It

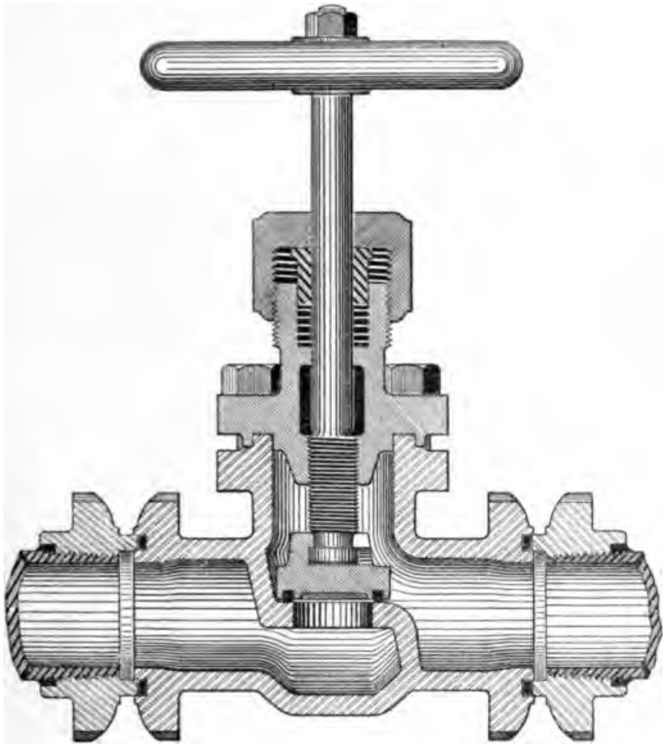


Fig. 37.

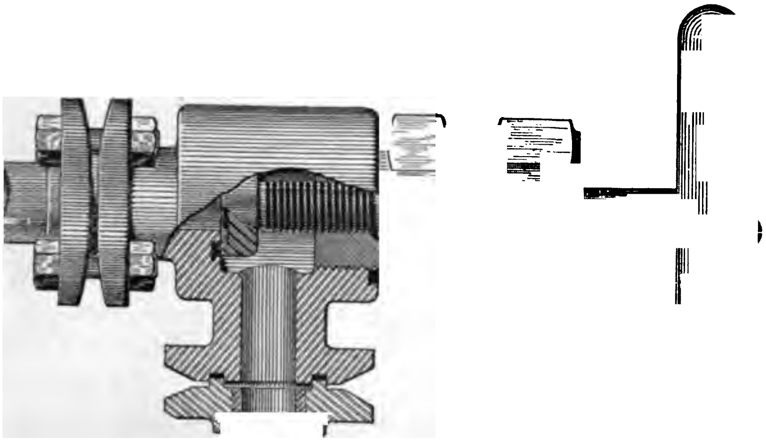


Fig. 38

may be well, however, to caution the user on the importance of testing the gauges often enough to be sure they are accurate, as serious damages may result from a wrong indication of pressure.

BRINE.

Brine is used in refrigeration as a medium for the transmission of heat from the point or object to be refrigerated to the ammonia or other refrigerant, and accordingly may be found in plants in which the application is indirect.

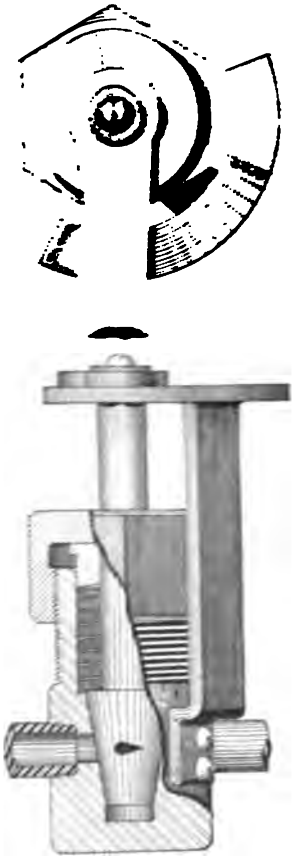
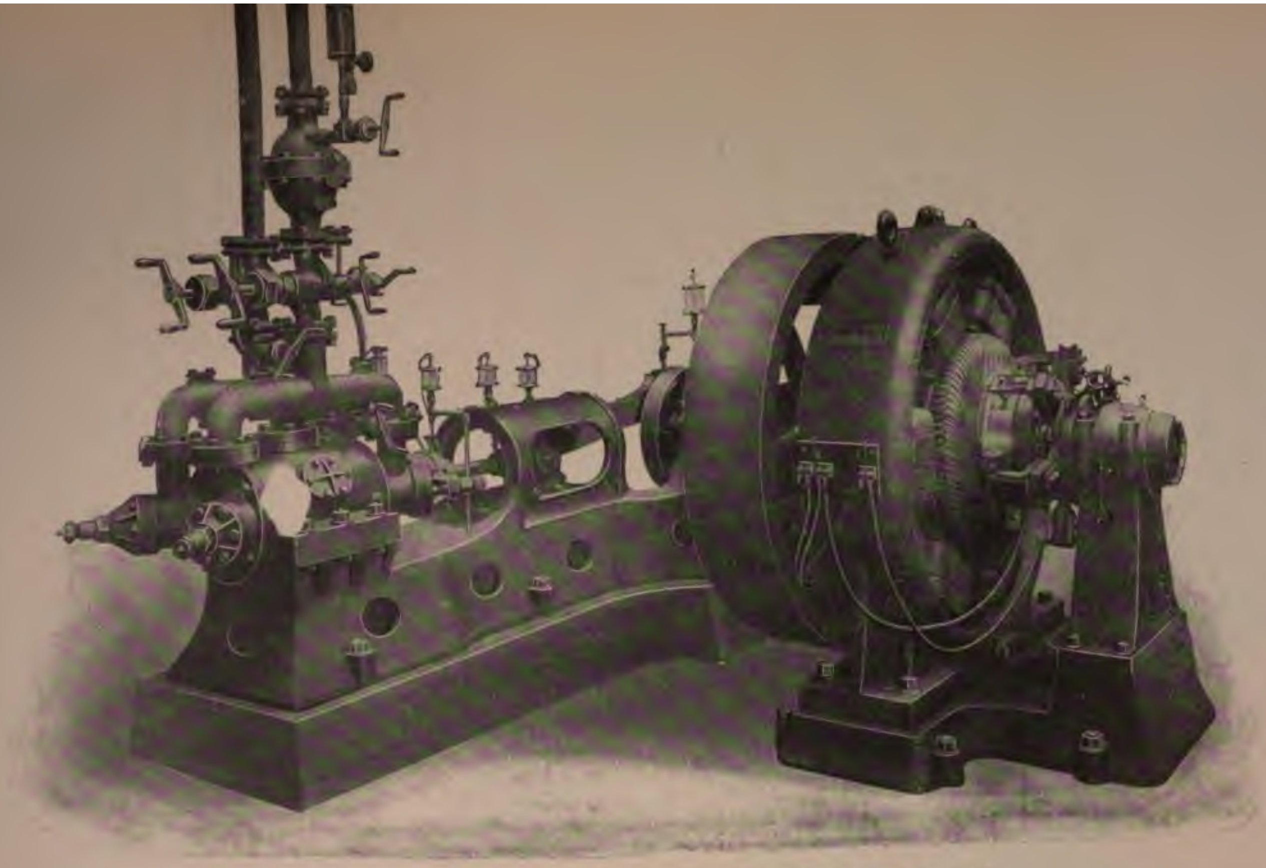


Fig. 39.

In the refrigerating practice of today, two kinds of brine are in use, chloride of calcium, and chloride of sodium (common salt). Either one is dissolved in water making a solution of the proper density; it is then pumped from the brine tank or cooler to the objects to be refrigerated, and returned by a system of pipes to be recooled and again circulated through the refrigerating system.

Chloride of calcium brine may be made of such density that its freezing point is −54° Fahr. (54° below zero) while chloride of sodium (salt) brine of maximum density or strength freezes at 0° Fahr. It will, therefore, be seen that for very low temperatures calcium brine should be used. For brine coolers in which the brine passes through the pipes, and the ammonia is evaporated on the outside with a temperature of a few degrees below zero, and the brine not in active circulation, salt brine would be frozen, bursting the pipes and causing a considerable loss. The calcium brine may be made of such density or strength as to make this impossible It will, therefore, be evident that for use in connection with a brine



DIRECT-CONNECTED TRIUMPH 10-TON REFRIGERATING MACHINE.

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cooler, the chloride of calcium brine is necessarily the choice It also has no corroding effect on pipes, pumps or other connections, and while the cost is somewhat more than that of salt, the general advantages are decidedly in its favor.

For the brine tank and coil system of refrigeration salt brine may be safely used, and is largely used at the present time. The evaporation of the ammonia, being within the pipes, can only effect the freezing of a small portion on the outer surface, and without damage to the plant. As its temperature may be reduced to zero, or even below (if circulation be maintained) it is effective for practical purposes wherein the temperature required is not below that point. The proper density, or strength of brine, either calcium or salt is determined by the temperature to which it is necessary to be reduced, and the tables on pages 79 and 80 will be found valuable in determining the proper strength for different requirements. It should be remembered, however, that a difference of from 5° to 10° Fahr. exists between the temperature of the brine and that of the evaporating ammonia, and that while the strength of the brine may appear ample for the temperature at which it is carried, the lower temperature of the evaporating ammonia may cause it to solidify within or upon the surface of the evaporator, thus causing it to separate, or freeze, and act as an insulator and prevent the transmission of heat through the surface. It is, therefore, necessary in examining into the strength of the brine, to consider it with reference to the evaporating pressure of the ammonia as well as its own temperature.

In the last column of the tables is given the gauge pressure, corresponding to the freezing point of the brine of different strengths.

The usual and proper instrument for determining the strength of brine is the Beaumè scale, for chloride of calcium brine. A weighted glass tube and bulb (Fig. 40) is graduated 0 to 100, the former being its floating point in water and the latter the floating point in a saturated solution of salt brine. By comparison of the two in the table of calcium brine it will be observed that the ratio of the two scales are as 1 to 4 which makes it possible to obtain one from the other.

In using the scales it is customary to draw a sample of the [illegible] glass test tube, raising its temperature to approximately [illegible] the scale: it will then float at the point on its scale corresponding with its strength.

Brine Systems. In the use of the foregoing tables for cal- [illegible] allowance should be made for the difference in [illegible]. The quantity required per gallon, or cubic [illegible] accordingly. The average quantity required [illegible] correspond with the tables.

[illegible] it is well to fit up a box with a perforated [illegible] readily obtained and equally effective [illegible] by taking a tight barrel or hogshead, into [illegible] four to six inches above the bottom [illegible] and which is bored with one-half inch holes. Over the false bottom lay a piece of coarse canvas or sack to prevent the salt falling through. A water connection is made in [illegible] side of the barrel near the bottom, between [illegible] head and the false bottom, and a [illegible] valve placed nearby to regulate [illegible] of water passing through. An [illegible] connection is made near the top of [illegible] its end so placed that the brine [illegible] the tank, and a wire screen [illegible] its end inside the cask with a [illegible] between it, and the opening, to [illegible]. See Fig. 41. The cask or [illegible] filled with the calcium or salt, [illegible] dissolves and overflows into the brine [illegible].

[illegible] test tube and Beaumé scale, or salom- [illegible] should be kept at hand, and frequent [illegible] that the strength may be regulated [illegible] or less rapidly. After the first [illegible] the mixer to remain in position for [illegible]. A [illegible] connection should be made from the [illegible] refrigerating system to the cask, with a controlling valve by the use of which the strength or density of the brine may be increased without adding to the quantity, keeping the cask full of the calcium or salt, and allowing a portion of the return flow of brine to pass through the cask, dissolving the contents and flowing into the tank.

Calcium is usually obtained in sheet-iron drums, holding about 600 pounds each; it is in the shape of a solid cake within the drum. It is advisable to roll these onto the floor or top of tank in which the brine is to be made and pounded with a sledge hammer before removing the iron casing, this process breaking it up into small pieces without its flying about the room. After breaking it up the shell may be taken off and the contents shoveled into the mixer.

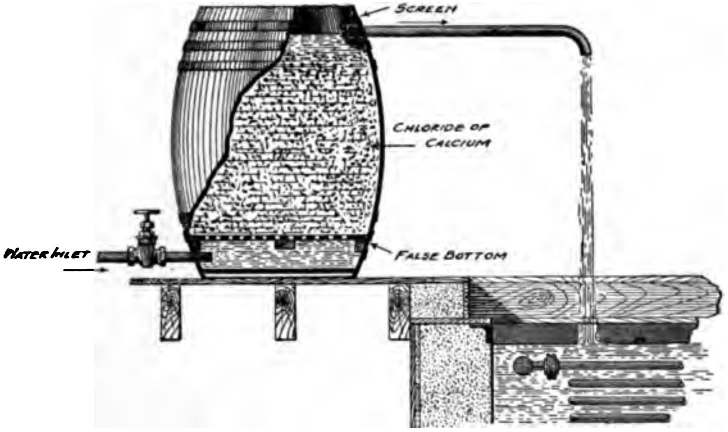

Fig. 41.

It is also sold and shipped in liquid form, in tank cars, generally in a concentrated form (on account of freight charges) and diluted to the proper point upon being put into the plant. Where proper railroad facilities exist, this is probably the most desirable way of obtaining the calcium.

Salt is sold and may be obtained in a number of forms. The usual shape for brine is the bulk, or in sacks of about 200 pounds each. Where it is possible to handle salt in bulk, direct from the car to the tank, this is most generally used on account of the price, being about $1.00 per ton less than if sacked. If it is necessary for it to be carted or stored before using, the sack form is preferable. The coarser grades of salt are used for this purpose, No. 2 Mine being the grade commonly used. The finer salts are higher in price, without a corresponding increase in strength of the brine formed.

The proper density or strength of the brine must be determined for various temperatures. As a rule its freezing point should be equal to, or slightly below, the temperature of the evaporating ammonia, rather than the temperature of the coldest brine, as is common. Referring to the table of salt brine solution, page 80; if we wish to carry a temperature of 10° Fahr. in the outgoing brine, it is necessary that the temperature of the evaporating ammonia be from 5° to 10° degrees below that point, in order that the transfer of heat from the brine to the ammonia will be rapid enough to be effective, which would mean that the ammonia would be evaporating at a temperature of practically 0° Fahr. To prevent the brine freezing against the walls of the evaporator its strength or density should be made to correspond with this, or from 95° to 100° on the salometer. This should be cared for more especially in connection with plants using the brine cooler, in which the brine is within the coil, than in the brine tank system in which the ammonia is inside the pipe and the brine outside; here the only danger or loss would be in efficiency.

In examining into the causes of failure in a plant to perform its usual or rated capacity, it is advisable, unless there is every evidence that the trouble is elsewhere, to make an examination of the brine and determine that its strength and condition is suited to the duty to be performed.

DIRECT EXPANSION.

As its name would imply, this system of refrigeration is one in which the refrigerant is expanded or evaporated in direct contact with the duty to be performed, without an intermediate agent for the transfer of the heat. Its application admits of quite a variety of apparatus to meet the requirements of refrigerating practice, the most general of which is the expansion within a pipe system placed in a room to be refrigerated, or within or between a series of pipes over or within which the substance to be refrigerated is passed.

While the former system admits of a variety of arrangements of the pipes within the room or chamber to be refrigerated, it is confined to the simple principle, however, of the evaporation of the refrigerant within the pipes, by the transfer of heat through the walls of the pipe, thereby reducing the temperature in the room or chamber to the desired point, and for certain purposes is a very satisfactory means of producing the desired results. This is principally true with large rooms in which the temperature and duty to be performed is constant, such as a brewery, packing house, and cold-storage rooms. For rooms requiring an unusually low temperature, as freezers of fish and poultry, direct expansion is desirable, because it is possible to more nearly reach the temperature of the refrigerant direct, than with an intermediate agent, as brine; this is due to the fact that there must necessarily be a difference of from 5° to 10° between the temperatures.

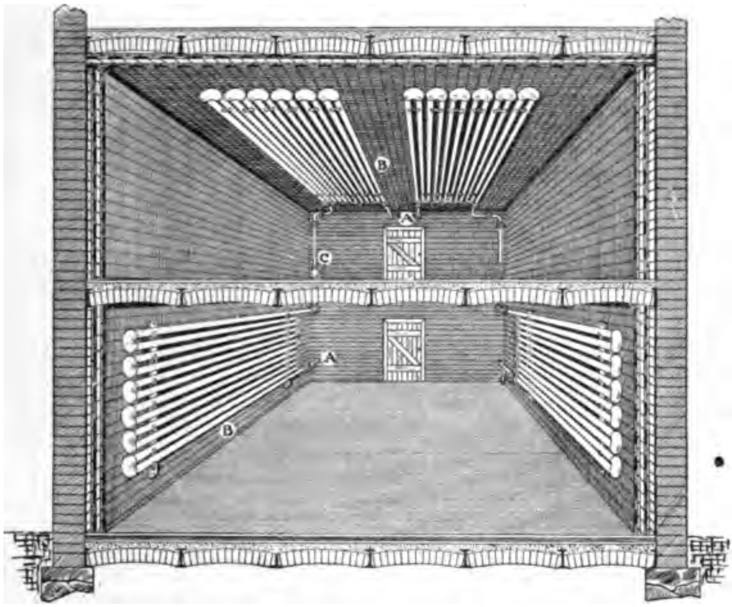

Fig. 42.

Fig. 42 illustrates a direct expansion construction in which the liquid anhydrous ammonia is expanded by the valve or cock A into the coil or system of pipes B and the gas returning to the compressor through the pipe C.

Fig. 43 represents a fish-freezing room on either side of which is arranged a series of pipe shelves, through which the ammonia is evaporated. The fish are laid on tin trays and placed on the pipe shelves until the room is filled. It is then closed, the ammonia turned on and left until the freezing has been accomplished.

It is apparent from the arrangement of the coils being both above and below the trays holding the fish, and close together, that the effect must be very rapid.

In the cooling of beer or other liquids, two forms of apparatus are used: one (the most common), being a series of a 2-inch pipe with return bends, stands, etc., over which the liquid to be cooled is flowed and within which the ammonia is evaporated, the product being accumulated in an iron pan or other receptacle in which the cooling pipes are placed. In brewery practice it is customary to provide a double series of coils, one above the other, cold water being circulated through the upper section, and the ammonia through the lower, the effect of which is first to remove the heat from the wort as far as possible by the use of water, and the remainder of the cooling being accomplished by the evaporation of the ammonia. This apparatus is called the Baudelot Cooler and is illustrated in Fig. 44.

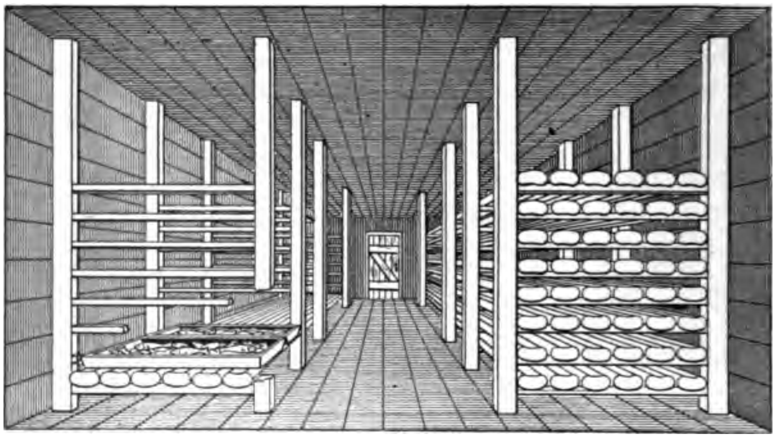

Fig. 43.

Fig. 45 represents a section of a double-pipe cooler for water or other liquids and is similar to the form of double-pipe condenser or cooler already described. It varies only in the construction of the inner tube which is corrugated to prevent bursting by freezing, which becomes possible when used to cool a congealable liquid. In the operation of this type of cooler, when used as an evaporator or direct expansion cooler, it is usual to reverse the two currents, by feeding or expanding the ammonia into the bottom of the coil the gas issuing from the top, while the liquid to be acted upon enters at the top of the coil and issues from the bottom. This type of cooler is largely used in cooling carbonated ale, as it must be kept from the atmosphere, and for cooling drinking water in circulating systems now installed in hotels, office buildings and department stores.

Purging and Pumping out Connection. A common cause of failure to operate properly and effectively is the introduction of some foreign substance into the system. This will be readily understood and appreciated by engineers and those familiar with the requirements of a steam boiler. Clean surfaces on the shell or tubes are necessary for the maximum evaporation of water, or the transfer of heat through the walls of pipe or other forms of heat-transmitting surface. The most common difficulty encountered in a refrigerating plant is oil, either in its natural condition, or saponified by contact with the ammonia, water or brine. It enters the system in many ways: through leakage, condensation in blowing out the coils or system, foreign gas arising from decomposition of the ammonia through excessive heat and pressure, or the mingling of air which may enter the system through pumping out below atmospheric pressure or the air may have remained in the system from the time of charging, never having been fully removed. It is also probable, though hard to determine with certainty owing to the various conditions surrounding the operation of plants, that impurities are introduced with the ammonia, either as a liquid, gas or air which afterwards becomes impossible to condense.

The oil in a system forms a covering or coating on the evaporating surface which acts as an insulation and prevents the ready transfer of heat through the walls of the evaporator. The presence of water or brine causes an absorption of a portion of the ammonia into the water or brine, forming aqua ammonia which raises the boiling point of the ammonia and causes material loss in the duty. Air or other non-condensable gas in the system, excludes an equal volume of the ammonia gas, thereby reducing the available condensing surface in that proportion.

For the purpose of cleaning the system and removing the different impurities which may appear, purge and blow-off valves and connections are provided. One of these is placed at or near the bottom of the oil interceptor, which is located between the compressors and the condenser; it is used to draw off the oil used as a lubricant in the compressor and precipitated to the bottom. This oil should not be allowed to accumulate to any great extent as it may be carried forward to the condenser by the current of the gas.

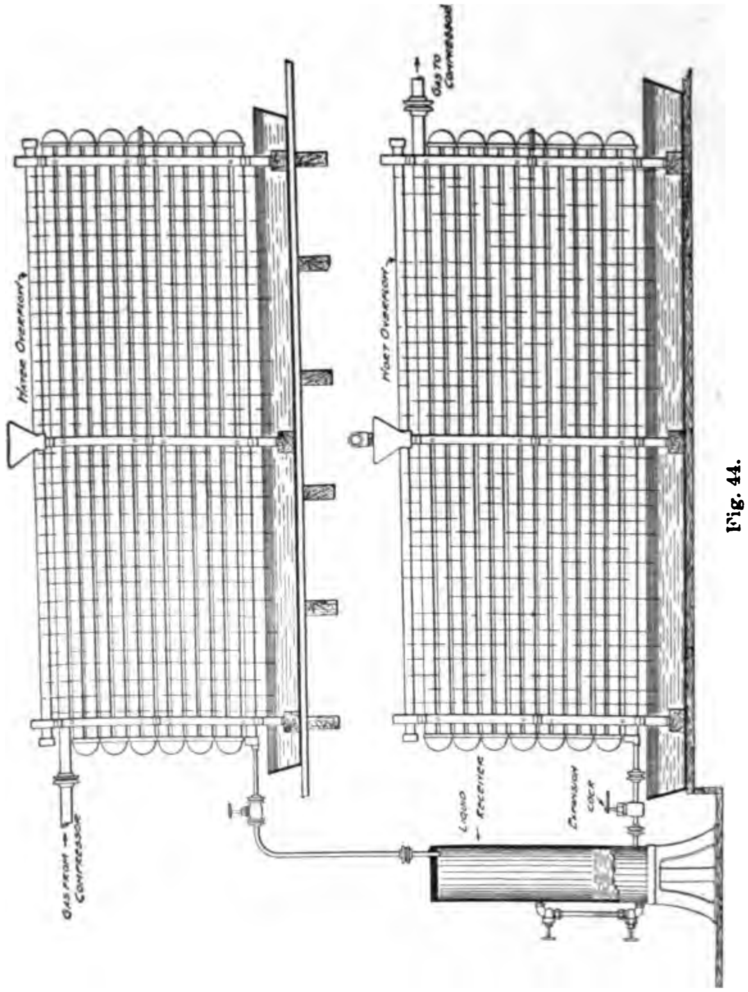

Fig. 44.

If the liquid ammonia receiver be placed in a vertical position it is customary to place a purge valve in the bottom for drawing off oil or other impurities. The supply of liquid to the evaporator being taken off at a short distance above the bottom say 4 to 6 inches.

The next point for the removal of impurities is at the bottom of the brine cooler, or lower manifold of the coil system in a brine-tank refrigerator. These may be tried as often as necessary to determine the state of cleanliness of the system. If the system is charged with any of the common impurities, they should be blown out and the system cleansed at the earliest possible moment, as they cause a decided loss.

Air or foreign gases accumulate in the condenser because the constant pumping out of the evaporating system tends to remove them from that part of the system to the condenser. This point, therefore, is the most natural place for their removal. For this purpose it is customary on the best condensers to place a header or manifold at the top at one end, and connect each of the sections or banks with a valved opening. A valve is also placed at each end of the header, and a connection from one end of this header made to the return gas line between the evaporator and the compressors. By closing the stop valves on the gas inlet and liquid outlet of any one of the sections and opening the purge or pumping-out line into the gas line to the compressors, the section or bank may be emptied of its contents for repairs or examination and then connected up and put into service without shutting down the plant, or a material loss of ammonia. For purging of air or gas, the valve should be closed between this header and the machine, and the valve on the opposite end opened to the atmosphere, the valves on each section opened slightly in turn while the foreign gases are expelled. This process should not be used while the compressor is in operation, as the discharge of the ammonia into the condenser would keep the gas churned to the extent that it would become impossible to remove the foul gases, without removing a considerable portion of the ammonia also.

For this reason it is customary before blowing off the condenser to stop the compressor and allow the water to flow over the condenser until it is thoroughly cooled. Sufficient time should

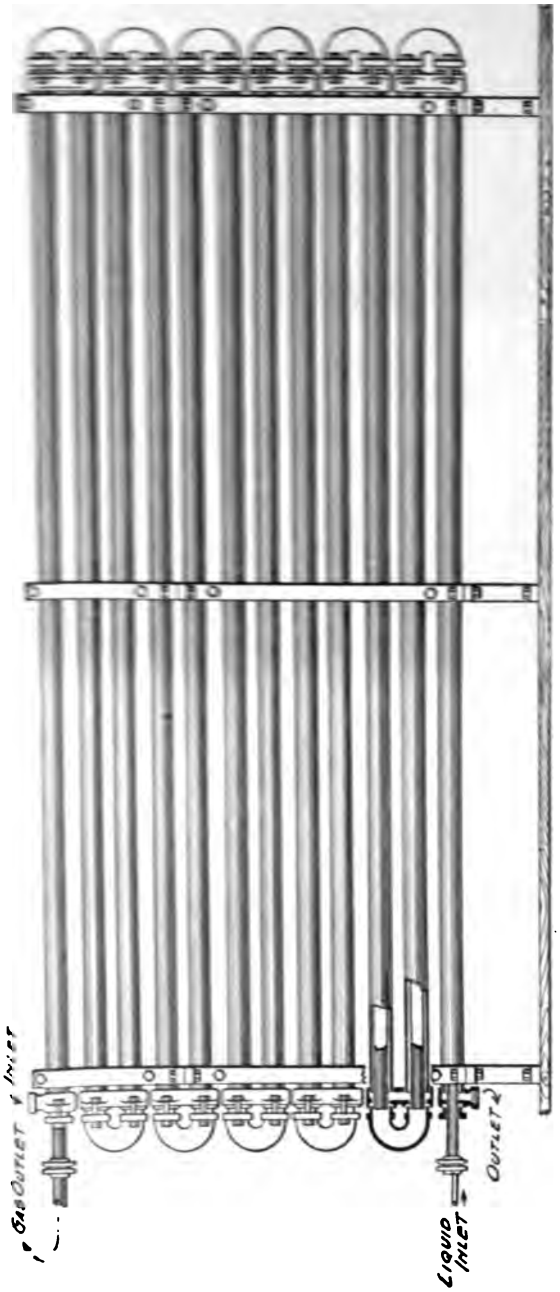

Fig. 45.

elapse for the ammonia to liquefy and settle towards the bottom, while the air and lighter gases rise to the top, at which point they may be blown out through the purge valve to the atmosphere. If doubt exists as to whether ammonia or impurities are being blown out attach a piece of hose to the end of the purge valve and immerse its other end in a pail of water. If it is air, bubbles will rise to the surface, while if it is ammonia, it will be absorbed into the water; the mingling of the ammonia with the water will cause a crackling sound, and the temperature of the water will increase owing to the chemical action.

TESTING AND CHARGING.

Having described the different parts of the refrigerating plant and their relations to one another, let us consider the process of testing and charging, or introducing the ammonia into the system. After the connections are made between the different parts, whether the system is brine or direct expansion, it is necessary to introduce air pressure into it to determine the state of the joints. This may be done in sections or altogether. It is customary, however, to put a higher pressure on the compression side of the plant than on the evaporator owing to the difference in the pressure carried in operation. Adjacent to each compressor is placed a main stop valve, on both the inlet and outlet sides, while on either side of these it is customary to place a by-pass or purge valve.

Before starting the compressor, the main stop valve or valves (if there be two) on the inlet or evaporating side of the compressor, is closed, the small valve between the compressor and the main stop valve opened, and all of the other valves on the system opened except those to the atmosphere. The compressor may then be started slowly, air being taken in through the small by-pass valves and compressed into the entire system. It is well to raise a few pounds pressure on the entire system, before admitting water into the compressor water jackets or other parts of the system, because if a joint were improperly made up, it would be possible for the water to enter the compressors, or coils of the condenser or evaporator, and serious damage or loss of efficiency in the plant occur which it might be impossible to locate afterwards. While if pressure exists within the system when the water is admitted, its

Many engineers require the vacuum test as well as the foregoing, and although if the former is gone over thoroughly, there can be little chance of leakage afterwards, it is better to be over-exacting than otherwise in the matter of testing and preparation of the plant, thus preventing the possibility of leaks that may prove disastrous. Open the main stop valves on the inlet line and close the main valves above the compressor on the discharge line, closing the by-pass valves in the suction line, and opening those in the discharge line between the main stop valve and the compressor. Have all the other valves on the system open as before for testing. Starting the compressor draws in the air filling the system through the compressor and discharging it at the small valve left open. Assuming the system to be tight, continuing the operation will finally exhaust the air (or nearly so), when the small valves should be closed and the pressure gauges watched to determine whether or not leakage exists.

Assuming that the system and apparatus is tight in every particular and that it is otherwise ready to be placed in operation, we are now ready to charge the ammonia into the plant.

If the air has been exhausted from the system in testing, this usual step need not be taken before charging, and it is only necessary to put the machine in proper condition to resume the pumping of the gas, and attach a cylinder of ammonia to the charging valve to enable the refrigeration to be commenced. The main stop valves above the compressor which were closed in expelling the air should now be opened and by-pass and other valves to the atmosphere closed. Close the outlet valve from the ammonia receiver and start the machine slowly, at the same time opening the feed valve between the drum of ammonia and evaporator. The anhydrous liquid ammonia will flow into the evaporator through the regular supply pipe, the gas resulting from evaporation being taken up by the compressors and discharged into the condenser and finally settling down into the receiver, when a sufficient quantity has been introduced to form a supply there. Upon closing the valves between the drum from which the supply is being drawn, and opening the outlet valve from the receiver, the process of refrigeration by the compression system is regularly in operation.

[illegible] MANAGEMENT OF THE PLANT

[illegible] necessary, and [illegible] of the pipes or other [illegible] operating system, and a proper [illegible]. It is also necessary that the [illegible] separated by contact with the [illegible] clogging the pipes, coating [illegible] causes a corresponding loss [illegible] purge valve in the bottom of the oil [illegible] about once each week, and the [illegible] drawn off into a pail or can, [illegible] provided. After the gas with which [illegible] charged has escaped, the oil should be practically the same as when fed into the compressor. If, however, the oil is not of the proper quality it will remain thick and pasty, or gummy, showing it to have been affected by the ammonia. Its use should not be continued.

By opening the purge valves, which are usually provided at the bottom manifold or header of the brine tank and the bottom head of the brine cooler, oil or water, if any be in that part of the system, may be drawn off. These valves, however, should not be opened unless there is some pressure in that part of the system, as air would be admitted if the pressure within the apparatus is below that of the atmosphere. Air may enter the system through a variety of causes and its presence is attended with higher condensing pressure and a falling off in the amount of work performed. For the removal of air from the apparatus a purge valve is placed at the highest point in the condenser or discharge pipe from the compressors near the condenser, which may be tried when the presence of air or foreign gases is suspected. This should be done after the compressor has been stopped. When the condenser has fully cooled and the gases separated, a small rubber hose or pipe may be carried into a pail of water and the purge valve or valves slightly opened. If air or other gases exist in the system bubbles will rise to the surface of the water so long as it is escaping, while, if ammonia is being blown off, it will be absorbed in the water and not rise to the surface.

To prevent the possibility of air getting into the system, the evaporating pressure should never be brought below that of the atmospheric, or 0° on the gauge, as at such times, with the least leakage at any point, it is sure to enter. Should it become necessary to reduce the pressure below that point, it is well first to tighten the compressor stuffing boxes and allow the pressure to remain below 0° only the shortest possible time as not only air may enter, but if it be the brine system and a leak exists, brine also will be drawn in.

From the foregoing it is evident that in order to obtain satisfactory results, the interior of the system must be kept clean by purging at the different points provided for this purpose; and it need only be added in this connection, that when the presence of oil or moisture becomes apparent in any quantity, the coils or other parts should be disconnected and blown out with steam until thoroughly clean and afterwards with air to make certain that condensation from the steam does not remain, after which the parts may again be connected and tested ready for operation.

If the plant be a brine system it is necessary that the brine be maintained at a proper strength or density to obtain satisfactory results, for if it becomes weakened it freezes on the surfaces of the pipes or evaporator, acting as an insulator and preventing the rapid transmission of heat through the walls.

Frequently the engineers or owners in looking about for the cause of a falling off in the capacity of a plant, overlook the fact that the strength of the brine is the sole difficulty, and the addition of salt or chloride of calcium is a remedy. Referring to the tables of brine solutions, there should be no difficulty in determining the proper strength of the brine for the different temperatures at which the plant may be operating; but as already stated, this should be made to correspond to the temperature of the evaporating ammonia rather than the temperature at which the brine may be handled.

It is of great importance to know at all times whether or not the gas taken into the compressors is fully discharged into the condenser, as the slightest loss at this point is certain to make itself felt in the operation of the plant. The compressor or valves seldom need be taken apart to determine their operation. The engineer should be able to discern the slightest change in temperature from the normal, when the compressors are working at their best, by placing the hand on the inlet and outlet pipes or the lower part of the compressors. Should the inlet pipe to one compressor be warmer than that to the other (of a pair), or the frost on the pipe from the evaporator reach nearer one compressor than the other, it is then certain that the one with the higher temperature, or from which the frost is farthest, is not working properly or doing as much duty as the other, and it is equally certain that some condition exists which prevents the complete filling and discharge of its contents; possibly it has more clearance or leaky valves.

The most common difficulties experienced with ammonia condensers are those of keeping the external surfaces clean and free from deposits, and preventing the accumulation of air or foreign gases within. Deposits on the surface are usually of two kinds, one

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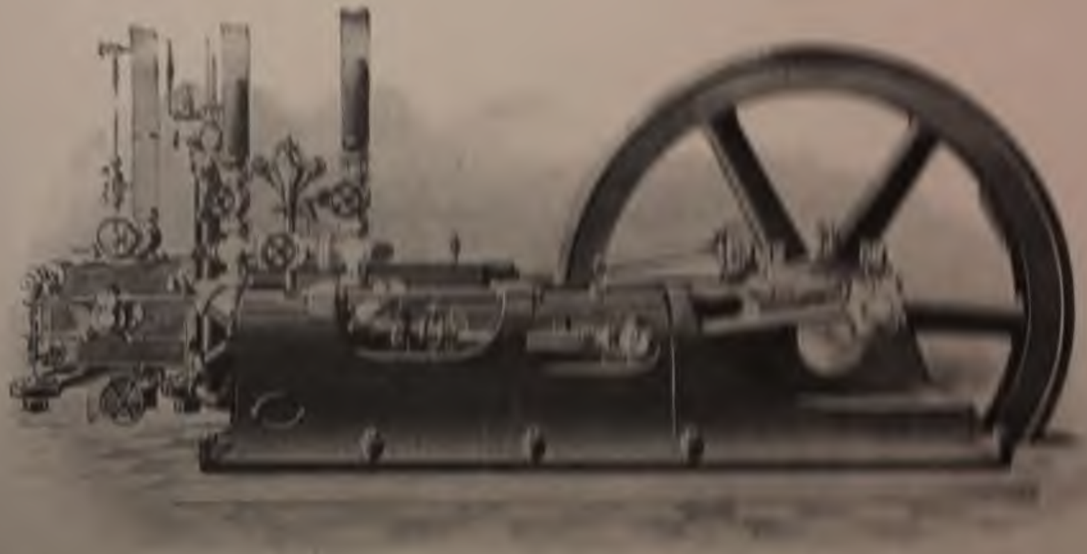

TANGYE FRAME HORIZONTAL LINDE COMPRESSOR.
Fred W. Wolf Company.

which remains soft and may be washed off with a brush or wire scraper such as is used for cleaning castings in a foundry, the other a hard deposit which must be loosened with a hammer or scraper. It is hardly necessary to explain in detail the methods employed in cleaning the condenser as this is a matter that each engineer will be able to accomplish in his own way. It should not, however, be overlooked, and with a condensing pressure higher than ordinary, this should be the first point to be examined after the water supply.

Air, or foreign gases due to decomposition of the ammonia or other causes find their way into the condenser and make themselves manifest generally in a higher condensing pressure, or a falling off in the duty to be obtained from the plant. They should be blown off through the purge valve at the top of the condenser in the manner already described.

It is possible, through leakage of the coils or other parts of the apparatus, that the ammonia may become mixed with brine or water, thereby retarding its evaporation and interfering with the proper or usual operation of the plant. If this is suspected, a sample may be drawn off into a test glass through the charging valve or purge valve of the brine tank or ammonia receiver, and allowed to evaporate, in which case the water or brine will remain in the glass and the relative amount be determined. Through careful evaporation, and continued purging of the evaporator at intervals, this may in time be eliminated, and care should be taken to prevent future recurrence.

Loss of ammonia should be constantly guarded against. It is watchfulness which determines between a wasteful and an economical plant in this particular, and the engineer who allows the slightest smell of ammonia to exist about the plant is certain to be confronted with excessive ammonia bills; while he who is constantly on the alert and never rests until his plant is as free from the smell of ammonia as an ordinary engine room, will be referred to as the one who ran such and such a plant without addition of more ammonia for so many years.

The escape of ammonia into the atmosphere is readily detected; but where a leakage occurs in a submerged condenser, brine tank or brine cooler it is necessary to examine the surrounding liquid to determine whether or not it exists. For this purpose [illegible] of druggists or [illegible] paper when [illegible] ammonia will turn [illegible] yellow [illegible] pink color with [illegible]

[illegible] in the brine tank or [illegible] operation, by shutting [illegible] is accessible a [illegible]

[illegible] BETWEEN THE PARTS OF A REFRIGERATING PLANT

[illegible] proportion between the [illegible] is between the [illegible] power plant, in order to [illegible] necessary that the [illegible] surface sufficient to [illegible] the ammonia, for [illegible] going into a [illegible] in both lineal [illegible] commercial sizes [illegible]

[illegible] refrigeration should [illegible] pipe surface, to [illegible] when it is to be [illegible] This is an ample allow[illegible] conditions to give [illegible] have been made [illegible] found sufficient for [illegible] to use the former amount [illegible] a plant may be ope[illegible] figures to 150 linear feet of [illegible] 100 feet of 1½-inch, or 80 feet of [illegible]

[illegible] contain from 40 to 60 cubic feet (depending upon the amount of storage capacity desired) for each ton in capacity of the plant. The brine cooler should contain 12 square feet of external pipe surface for each ton, from which, in comparison with the amount required for the brine-tank system, the statements regarding the relative efficiency of the two methods of cooling brine may be more readily understood. This amount of surface would practically correspond to 35 linear feet of 1-inch pipe, 28 feet of 1¼-inch, 25 feet of 1½-inch or 19 feet of 2-inch pipe. The shell of the cooler is made sufficiently large to contain only the number of coils necessary, there being no advantage in a larger shell.

The submerged type of ammonia condenser should contain approximately 35 square feet of external surface which nearly corresponds to 100 linear feet of 1-inch pipe, 80 feet of 1¼-inch, 70 feet of 1½-inch and 56 feet of 2-inch pipe.

The atmospheric type of condenser should contain 30 square feet of external pipe surface which corresponds to 87 linear feet of 1-inch pipe, 69 feet of 1¼-inch, 60 feet of 1½-inch and 48 feet of 2-inch pipe.

The double-pipe type of condenser, as usually rated, contains 7 square feet of external pipe surface for the water circulating pipe and about 10 square feet of internal surface of the outer pipe and corresponds to approximately 20 linear feet each of 1¼-inch and 2-inch sizes for each ton of refrigerating capacity.

The above quantities are based on a water supply of average temperature (60°) and quantity. In cases of a limited supply or higher temperature than ordinary, a greater amount should be used.

The ammonia compressor should be of such dimensions that it will take away the gas from the brine cooler, evaporating coils, or system, as rapidly as formed by the evaporation of the liquid ammonia, and unless the temperature at which the plant is to be operated be known, it is impossible to determine the volume of gas to be handled and the necessary sizes of the compressor.

As stated before, the unit of a refrigerating plant is usually expressed in tons of refrigeration equal to 284,000 B. T. U. Up to the present time, however, a standard temperature at which this duty shall be performed has never been established, and therefore the rating of a machine, evaporator, or condenser by tonnage is a merely nominal one and misleading to the purchaser, a range of as great as 50 per cent very often existing in the tenders for certain contracts. Upon the basis, however, of the average temperature required of the refrigerating apparatus that of 15° Fahr. is probably the mean; and at this temperature in the outgoing brine, it is necessary to take away from the evaporator nearly 7,000 cubic inches of gas per minute for each ton of refrigeration developed in twenty-four hours. This may be considered as a fair basis for the rating of the displacement of the compressor or compressors of the plant, unless a specific temperature is stated at which the plant is to operate. At 0° Fahr. it is necessary to calculate on approximately 9,000 cubic inches, while at 28° Fahr. about 5,000 will be the required amount.

For example, if we have two single acting compressors 12 inches diameter by 24 inches stroke operating at 70 revolutions per minute, we would have 113.09 (inches, area of 12-inch circle) × 24 (inches stroke) × 2 (number of compressors) × 70 (revolutions) ÷ 7,000 (cubic inches displacement required) = 54.28 (tons refrigeration per 24 hours of operation), while if the same machine is to be operated at or near a temperature of zero and we divide the product by 9,000 we have a capacity of 42.22 tons only in the same length of time. The above quantities are given as approximate only, but they have been deduced from the average results obtained from years of practice and will be found reliable under average conditions. It is to be hoped, however, that a standard will soon be adopted which will rate machines or plants by cubic inches displacement at a certain number of revolutions or a stated piston speed, and the cooling of a certain number of gallons of brine per minute through a certain range of temperature.

THERMOMETER SCALES.

Fahr.	Cent.	Reau.	Fahr.	Cent.	Reau.	Fahr.	Cent.	Reau.
212	100	80	120	48.9	39.1	30	— 1.1	— 0.9
210	98.9	79.1	118	47.8	38.2	28	— 2.2	— 1.8
208	97.8	78.2	116	46.7	37.3	26	— 3.3	— 2.7
206	96.7	77.3	114	45.6	36.4	24	— 4.4	— 3.6
204	95.6	76.4	112	44.4	35.6	22	— 5.6	— 4.4
202	94.4	75.6	110	43.3	34.7	20	— 6 7	— 5.3
200	93.3	74.7	108	42.2	33.8	18	— 7.8	— 6.2
198	92.2	73.8	106	41.1	32.9	16	— 8.9	— 7.1
196	91.1	72.9	104	40	32	14	—10	— 8
194	90	72	102	38.9	31.1	12	—11.1	— 8 9
192	88.9	71.1	100	37.8	30.2	10	—12.2	— 9 8
190	87.8	70.2	98	36.7	29.3	8	—13.3	—10.7
188	86.7	69.3	96	35.6	28.4	6	—44.4	—11.6
186	85.6	68.4	94	34.4	27.6	4	—15.6	—12.4
184	84.4	67.6	92	33.3	26.7	2	—16 7	—13.3
182	83.3	66.7	90	32.2	25.8	0	—17.8	—14 2
180	82.2	65.8	88	31.1	24.9	— 2	—18.9	—15.1
178	81.1	64.9	86	30	24	— 4	—20	—16
176	80	64	84	28.9	23.1	— 6	—21.1	—16.9
174	78.9	63.1	82	27.8	22.2	— 8	—22.2	—17.8
172	77.8	62.2	80	26 7	21.3	—10	—23.3	—18.7
170	76.7	61.3	78	25.6	20.4	—12	—24.4	—19.6
168	75.6	60.4	76	24.4	19.6	—14	—25.6	—20.4
166	74 4	59.6	74	23.3	18.7	—16	—26.7	—21.3
164	73.3	58.7	72	22.2	17.8	—18	—27.8	—22.2
162	72.2	57.8	70	21.1	16.9	—20	—28.9	—23.1
160	71.1	56.9	68	20	16	—22	—30	—24
158	70	56	66	18.9	15.1	—24	—31.1	—24.9
156	68.9	55.1	64	17.8	14.2	—26	—32.2	—25.8
154	67.8	54.2	62	16.7	13.3	—28	—33.3	—26.7
152	66.7	53.3	60	15.6	12.4	—30	—34.4	—27.6
150	65 6	52.4	58	14.4	11.6	—32	—35.6	—28.4
148	64 4	51.6	56	13.3	10.7	—34	—36.7	—29.3
146	63.3	50.7	54	12.2	9.8	—36	—37.8	—30.2
144	62.2	49.8	52	11.1	8.9	—38	—38.9	—31.1
142	61.1	48.9	50	10	8	—40	—40	—32
140	60	48	48	8.9	7.1	—42	—41 1	—32.9
138	58.9	47.1	46	7.8	6.2	—44	—42.2	—33.8
136	57.8	46.2	44	6.7	5.3	—46	—43.3	—34 7
134	56.7	45.3	42	5.6	4.4	—48	—44.4	—35.6
132	55.6	44.4	40	4.4	3.6	—50	—45.6	—36.4
130	54.4	43.6	38	3.3	2.7	—52	—46.7	—37 3
128	53.3	42.7	36	2.2	1.8	—54	—47.8	—38.2
126	52.2	41.8	34	1.1	0.9	—56	—48.9	—39.1
124	51.1	40.9	32	0	0	—58	—50	—40
122	50	40						

Temperature, Pressure, Latent Heat and Weight of Ammonia, or Properties of Saturated Ammonia.

Temp. F.	Pressure		Latent Heat	Volume in Cu. Ft. of 1 Lb. Vapor	Volume in Cu. Ft. of 1 Lb. Liquid	Weight of 1 Cu. Ft. Vapor
	Absolute	Gauge				
—40	10.69	—4 01	579.67	24.88	.0234	.0411
—35	12.31	—2.39	576 69	21.82	.0236	.0471
—30	14.13	— .57	573.69	18.69	.0237	.0535
—25	16.17	1.47	570.68	16.44	.0238	.0609
—20	18.45	3.75	567.67	14 48	.024	.069
—15	20.99	6.29	564.64	12.81	.0242	.0775
—10	23.77	9.07	561 61	11.36	.0243	.088
— 5	27.57	12.87	558.56	9.89	.0244	.1011
0	30.37	15.67	555 5	9.14	.0246	.1094
+ 5	34.17	19.47	552.43	8.04	.0247	.1243
10	38.55	23.85	549.85	7.2	.0249	.1381
15	42.93	28.23	546.26	6.46	.025	.1547
20	47.95	33.25	543 15	5.82	.0252	.1721
25	53.43	38.73	540.03	5.24	.0253	.1908
30	59.41	44.71	536.92	4.73	.0254	.2111
35	65.93	51.23	533.78	4.28	.0256	.2336
40	73	58.3	530.63	3.88	.0257	.2577
45	80.66	65.96	527.47	3.53	.026	.2832
50	88.96	74.26	524.3	3.21	.02601	.3115
55	97 63	82.93	521.12	2.93	.02603	.3412
60	107.6	92.9	517 93	2.67	.0265	.3745
65	118.03	103.33	515.33	2.45	.0266	.4081
70	129.21	114.51	511.52	2 24	.0268	.4664
75	144.25	126 55	508.29	2.05	.027	.4978
80	154.11	139.41	504 66	1.89	.0272	.5291
85	167.86	153.16	501.81	1.74	.0273	.5747
90	182.8	168.1	498.11	1.61	.0274	.6211
95	198.37	183.67	495 29	1.48	.0277	.6756
100	215.14	200.44	491.5	1.36	.0279	.7353
105	232.98	218.28	488.72	1.29	.0281	.7862
110	251.97	237.27	485.42	1.2	.0283	.8451
115	272.14	257.44	482.41	1.12	.0285	.9042
120	293.49	278.79	478 79	1.04	.0287	.9738
125	316.16	301.46	475.45	.97	.0289	1.172
130	340.42	325.72	472.11	.9	.0291	1.2218
135	365.16	350.46	468.75	.84	.0293	1.3212
140	392.22	377.52	465.89	.79	.0295	1.4108
145	420.49	405.79	462.01	.74	.0297	1.4904
150	450.2	435.5	458.62	.69	.0299	1.5896

TABLE OF CALCIUM BRINE SOLUTION.

Deg. Baumé 60° F.	Deg. Salometer. 60° F.	Per Cent Calcium by Weight	Lbs. per Cu. Ft. Sol.	Lbs. per Gallon	Specific Gravity	Specific Heat	Freezing Point F.	Amm. Gauge Pressure
0	0	0	0	0	1	1	32	47.31
1	4	.943	1.25	$\frac{1}{6}$	1.007	.996	31.1	46.14
2	8	1.886	2.5	$\frac{1}{3}$	1.014	.988	30.33	45.14
3	12	2.829	3 75	$\frac{1}{2}$	1.021	.98	29.48	44.06
4	16	3.772	5	$\frac{2}{3}$	1.028	.972	28.58	43
5	20	4.715	6.25	$\frac{5}{6}$	1.036	.964	27.82	42.08
6	24	5.658	7.5	1	1.043	.955	27.05	41.17
7	28	6.601	8.75	$1\frac{1}{6}$	1.051	.946	26.28	40.25
8	32	7.544	10	$1\frac{1}{3}$	1.058	.936	25.52	39.35
9	36	8.487	11 25	$1\frac{1}{2}$	1.066	.925	24.26	37.9
10	40	9.43	12 5	$1\frac{2}{3}$	1.074	.911	22.8	36.3
11	44	10.373	13.75	$1\frac{5}{6}$	1.082	.896	21.3	34.67
12	48	11.316	15	2	1.09	.89	19.7	32.93
13	52	12.259	16.25	$2\frac{1}{6}$	1.098	.884	18.1	31.33
14	56	13.202	17.5	$2\frac{1}{3}$	1.107	.878	16.61	29.63
15	60	14.145	18.75	$2\frac{1}{2}$	1.115	.872	15.14	28.35
16	64	15.088	20	$2\frac{2}{3}$	1.124	.866	13.67	27.04
17	68	16.031	21.25	$2\frac{5}{6}$	1.133	.86	12.2	25.76
18	72	16.974	22.5	3	1.142	.854	10	23.85
19	76	17.917	23.75	$3\frac{1}{3}$	1.151	.849	7.5	21.8
20	80	18.86	25	$3\frac{1}{3}$	1.16	.844	4.6	19.43
21	84	19.803	26.25	$3\frac{1}{2}$	1.169	.839	1.7	17.06
22	88	20.746	27.5	$3\frac{2}{3}$	1.179	.834	— 1.4	14.7
23	92	21.689	28.75	$3\frac{5}{6}$	1.188	.825	— 4.9	12.2
24	96	22.632	30	4	1.198	.817	— 8.6	9.96
25	100	23.575	31.25	$4\frac{1}{6}$	1.208	.808	—11.6	8.19
26		24.518	32.5	$4\frac{1}{3}$	1.218	.799	—17.1	5.22
27		25.461	33.75	$4\frac{1}{2}$	1.229	.79	—21.8	2.94
28		26.404	35	$4\frac{2}{3}$	1.239	.778	—27.	.65
29		27.847	36.25	$4\frac{5}{6}$	1.25	.769	—32.6	1″ Vac
30		28.29	37.5	5	1.261	.757	—39.2	8.5″ "
31		29 233	38.75	$5\frac{1}{6}$	1.272		—46.3	12
32		30.176	40	$5\frac{1}{3}$	1.283		—54.4	15
33		31.119	41.25	$5\frac{1}{2}$	1.295		—52.5	10
34		32.062	42.5	$5\frac{2}{3}$	1.306		—39.2	4
35		33	43.75	$5\frac{5}{6}$	1.318		—25.2	1.5

TABLE OF CHLORIDE OF SODIUM (SALT) BRINE.

Degrees on Salom.	Percentage Salt by Weight	Pounds Salt per Cu. Ft.	Pounds Salt per Gallon	Specific Gravity	Specific Heat	Freezing Point F.	Ammonia Gauge Pressure
0	0	0	0	1	1	32	47.32
5	1.25	.785	.105	1.009	.99	30.3	45.1
10	2.5	1.586	.212	1.0181	.98	28.6	43.03
15	3.75	2.401	.321	1.0271	.97	26.9	41
20	5	3.239	.433	1.0362	.96	25.2	38.96
25	6.25	4.099	.548	1.0455	.943	23.6	37.19
30	7.5	4.967	.664	1.0547	.926	22	35.44
35	8.75	5.834	.78	1.064	.909	20.4	33.69
40	10	6.709	.897	1.0733	.892	18.7	31.93
45	11.25	7.622	1.019	1.0828	.883	17.1	30.33
50	12.5	8.542	1.142	1.0923	.874	15.5	28.73
55	13.75	9.462	1.265	1.1018	.864	13.9	27.24
60	15	10.389	1.389	1.1114	.855	12.2	25.76
65	16.25	11.384	1.522	1.1213	.848	10.7	24.46
70	17.5	12.387	1.656	1.1312	.842	9.2	23.16
75	18.75	13.396	1.791	1.1411	.835	7.7	21.82
80	20	14.421	1.928	1.1511	.829	6.1	20.45
85	21.25	15.461	2.067	1.1614	.818	4.6	19.16
90	22.5	16.508	2.207	1.1717	.806	3.1	18.2
95	23.75	17.555	2.347	1.182	.795	1.6	16.88
100	25	18.62	2.488	1.1923	.783	0	15.67

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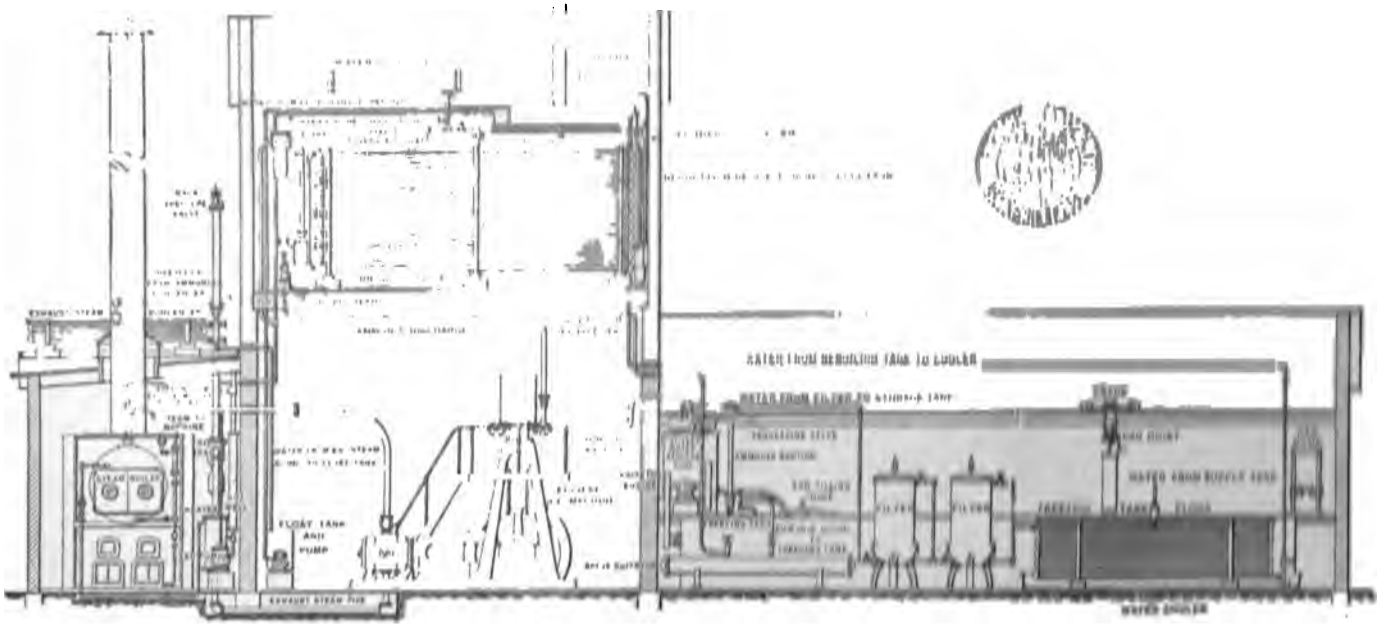

COMPLETE ICE FACTORY.
Frick Company.

REFRIGERATION.

PART II.

REFRIGERATION BY CARBONIC ANHYDRIDE GAS.

In general design and requirements, the refrigerating machine using carbonic anhydride gas is substantially like the machine using ether, ammonia, or carbonic dioxide gases in that it liquefies the gas by the removal of the latent heat in condensers and tanks. Where it differs materially is in the pressures required to accomplish the liquefaction of the gas.

The following table shows the pressure required for liquefaction at different temperatures from an approximate normal condition of 80 degrees to a point where the gas remains a liquid under the atmospheric pressure of 14.7 pounds from the absolute.

TEMPERATURE OF GAS LIQUEFIED.

Temperature Fahrenheit.	Pressure Pounds per sq. in.	Temperature Fahrenheit.	Pressure Pounds per sq. in.
80	1000.6	-60	134.2
70	889.3	-70	113.2
60	771.7	-80	96.2
50	689.4	-90	81.0
40	588.0	-100	70.1
30	507.1	-110	57.9
20	433.6	-120	47.4
10	367 1	-130	38.4
0	311.6	-140	30.0
-10	260.1	-150	21.3
-20	217.5	-160	15 2
-30	182 8	-170	7.1
-40	162.8	-179.6	0.0
-50	156.8		

In the operation of refrigerating machines where the gas is rapidly passing through the condensers, it has been established that the temperature of the gas liquefied is from 8 to 12 degrees above that of the water used for condensing. If the condensing water is 60 degrees, though during the summer days of August

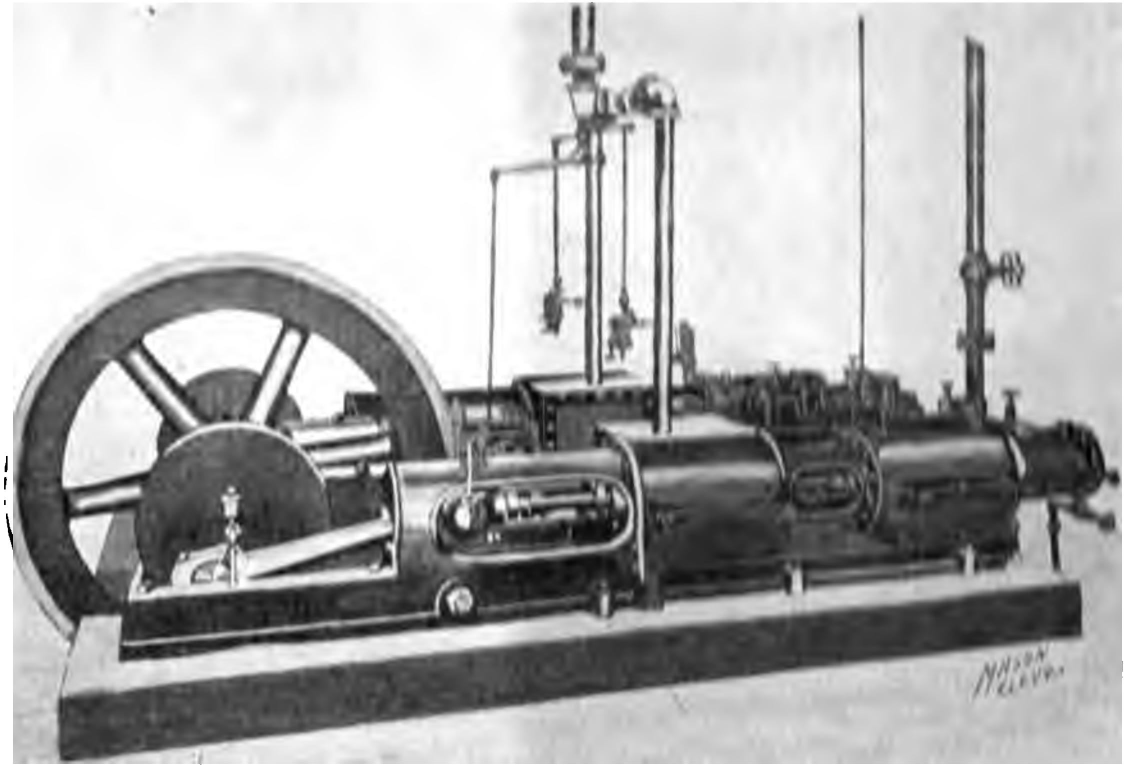

FIG. 46.

or September a temperature of 65 to 76 degrees is more common where city service water is in use, a pressure of at least 1,000 pounds per square inch is required for the working conditions.

With this pressure, compressing cylinders and piping must be very strong to withstand the pressure, and the diameter of the gas cylinder must be small compared with that of the steam cylinder. As carbonic anhydride gas does not attack copper or kindred metals they can be used with safety for the working parts of the machine.

Although as dangerous as any of the other gases, as it excludes the presence of oxygen, the gas is not noticed by smell as is the case with ammonia. In a vapor form, the specific gravity is greater than that of the atmosphere, and the gas escaping in small quantities is not easily detected as it immediately sinks in the cellars, sewers, or loose earth. As life cannot be sustained without the presence of oxygen, any gas that excludes it is dangerous. The presence of ammonia is so pronounced that it is easily detected when but a small amount is in the atmosphere, while the presence of carbonic anhydride gas is difficult to detect even up to the moment of exhaustion of life.

In the use of carbonic anhydride gas it is necessary that the most careful attention be paid to the joints and stuffing boxes. The compressors are invariably of the single-acting type, as it is almost impossible to maintain tight stuffing boxes under the intense pressure necessary for liquefaction

Compressors are quite extensively made for the liquefaction of carbonic acid anhydride gas, by industries supplying the same in small carboys of steel for recharging soda water or beer, or any beverage which requires the addition of carbonic acid gas for enlivenment. In the better class of these compressors, the cylinder for compressing the gas is compounded as shown in Fig. 46. By this design the first set of cylinders deliver the gas to a receiver at a pressure between 400 and 600 pounds per square inch. Here the gas is cooled; from this cylinder the suction of the final pump is taken, which leaves the gas at 1,000 pounds pressure. There is an advantage in machines of this build (compound cylinders), as there is a less amount of engine friction than in single-acting cylinders.

Fig. 47 shows a single-acting horizontal compressing cylinder direct connected to a throttling steam engine "D" valve pattern,

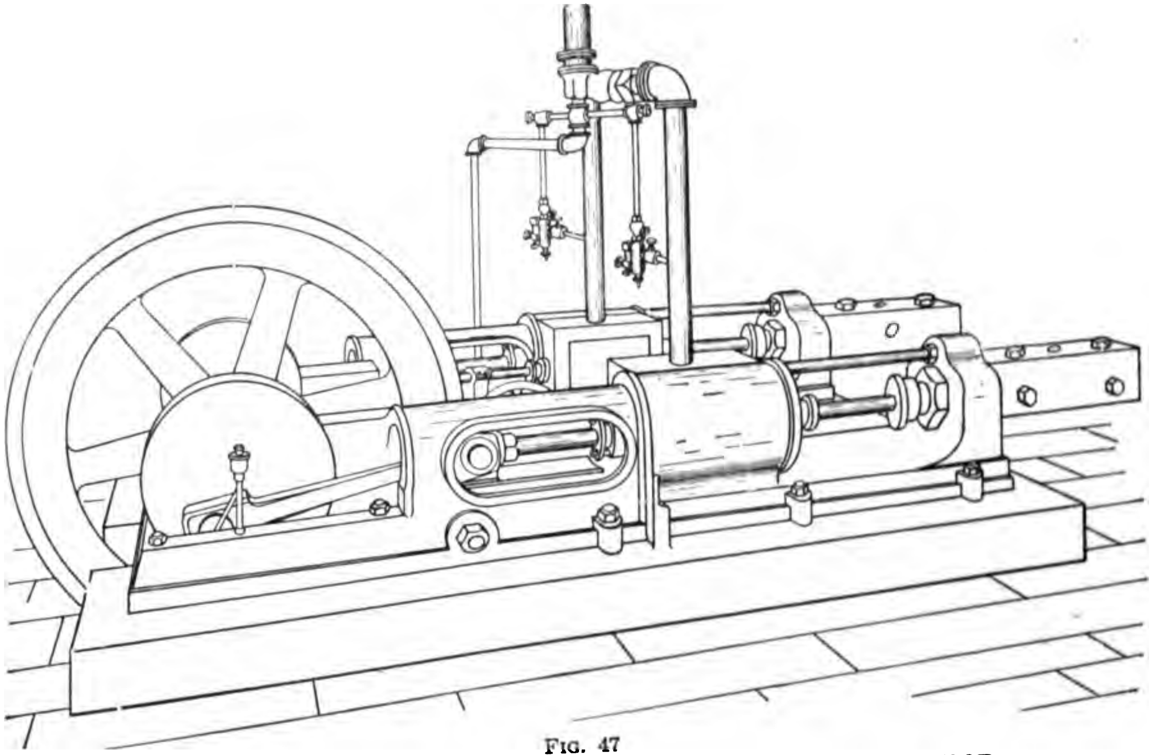

Fig. 47

DIRECT-CONNECTED HORIZONTAL REFRIGERATING COMPRESSOR

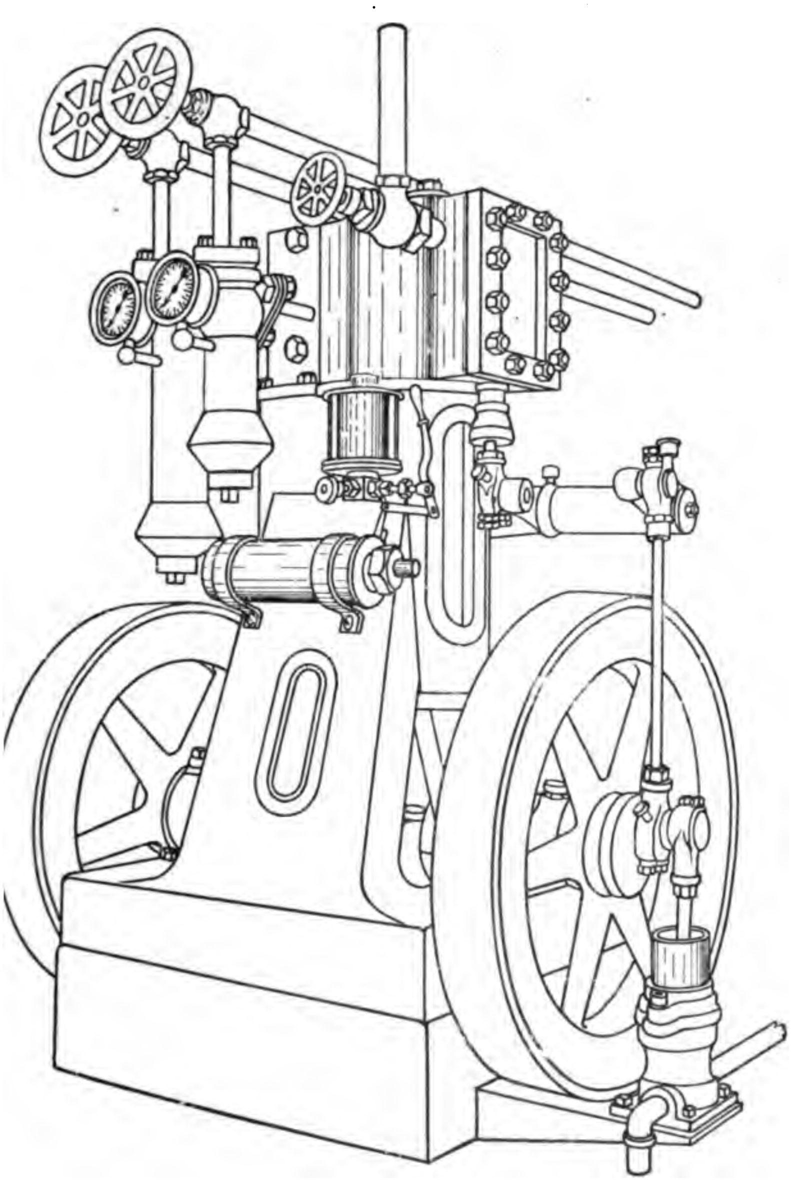

FIG. 48

IRECT-CONNECTED VERTICAL REFRIGERATING COMPRESSOR double cylinder operating on a single shaft. The tail rod of the steam cylinder extends into and forms the piston rod of the compressor.

Fig. 48 shows a vertical steam engine connected to a single-acting compressor; the compressor being bolted to the side of the steam cylinder with a double center-crank engine and over-hung fly wheel.

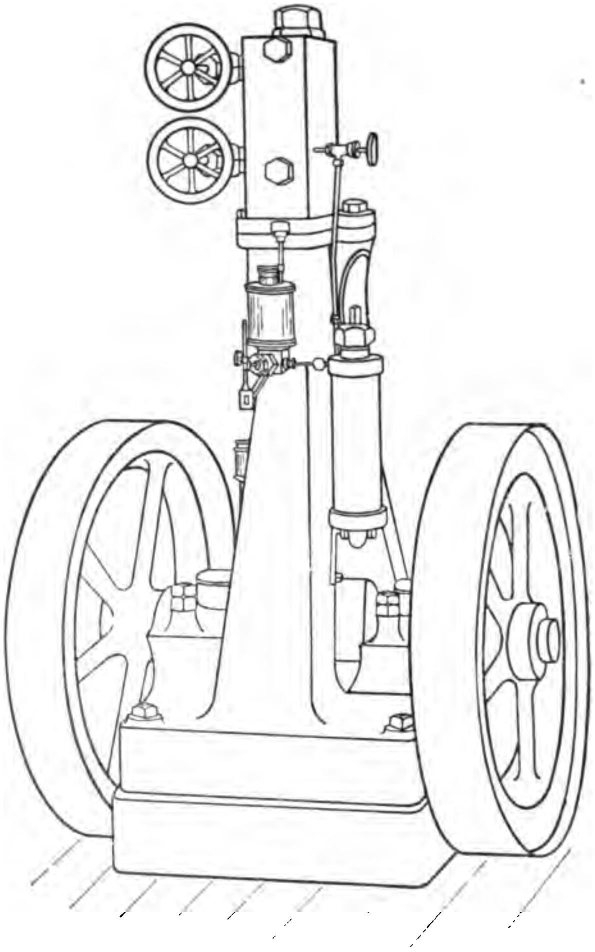

FIG. 49
BELT-DRIVEN REFRIGERATING COMPRESSOR

Fig. 49 shows a single-acting vertical compressor, connected

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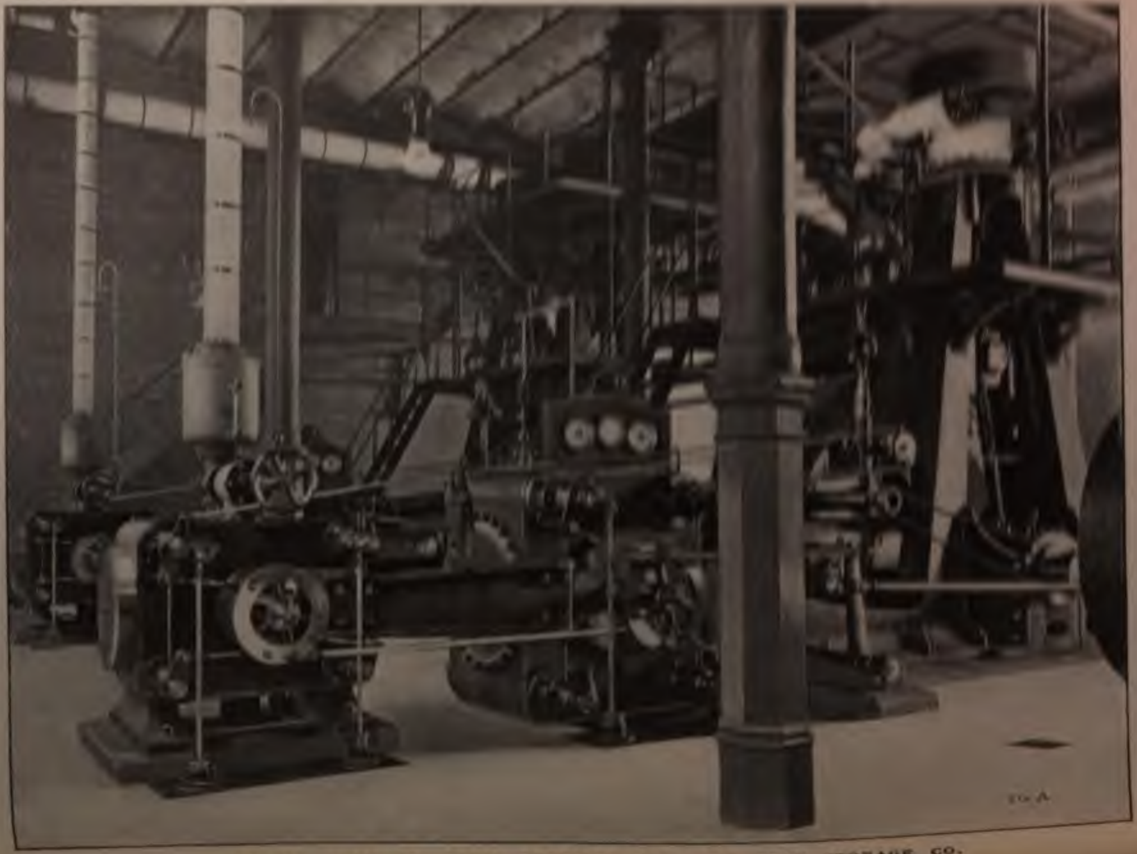


ENGINE ROOM ST. LOUIS REFRIGERATING AND COLD STORAGE CO.

to a shaft and belt, wheels over hanging. This design is for a belt-driven plant.

The cycle of operation of the gas is as follows: the gas passes from the compressor into a cylinder in which the oil from the lubricator of the cylinder is separated from the gas. From this tank the gas passes to the condensing coils where the latent heat is taken from it and the gas as it becomes a liquid flows to a receiving tank, being carried there by gravity.

From this liquid tank, the gas is conducted through piping to the freezing coils, where it is liberated from the condensing pressure to the vaporizing pressure, usually 30 to 50 pounds per square inch. A far greater back pressure can be carried than with ammonia when accomplishing the same work. This is due to the much lower vaporizing temperature, (boiling point) of the carbonic anhydride gas in a liquefied form when passing from a liquid to a vapor by the taking up of latent heat.

Carbonic Anhydride gas when liquefied to be used as a refrigerant may be applied through a brine tank, cooling the brine which afterwards may be circulated in cooling coils in refrigerator boxes or rooms. This system is known as the *indirect.* The process of sending the gas direct through the piping suspended in the refrigerating rooms is known as the direct application of refrigerating fluid.

REFRIGERATION BY COMPRESSED AIR.

To obtain refrigeration by compressing and expanding air, we resort to the effort of compression, and while air is compressed conduct off the heat which has multiplied in proportion to the amount of compression. We can convert the heat held by the air into power; then the air containing but little heat is used for refrigerating purposes.

That this may be more clearly understood let us consider the following: By taking air at atmospheric pressure at sea level, usually 14.7 pounds per square inch (this varies as the pressure increases or diminishes as shown by the mercury column in a barometer) and compressing it to 150 pounds per square inch, ten volumes of air have been compressed into the space occupied by one volume at atmospheric pressure. The heat which was contained in the ten volumes has been forced into the one volume.

The temperature is increased because the total amount of heat [illegible] of heat is in proportion to [illegible] factor which makes possible refrigeration [illegible].

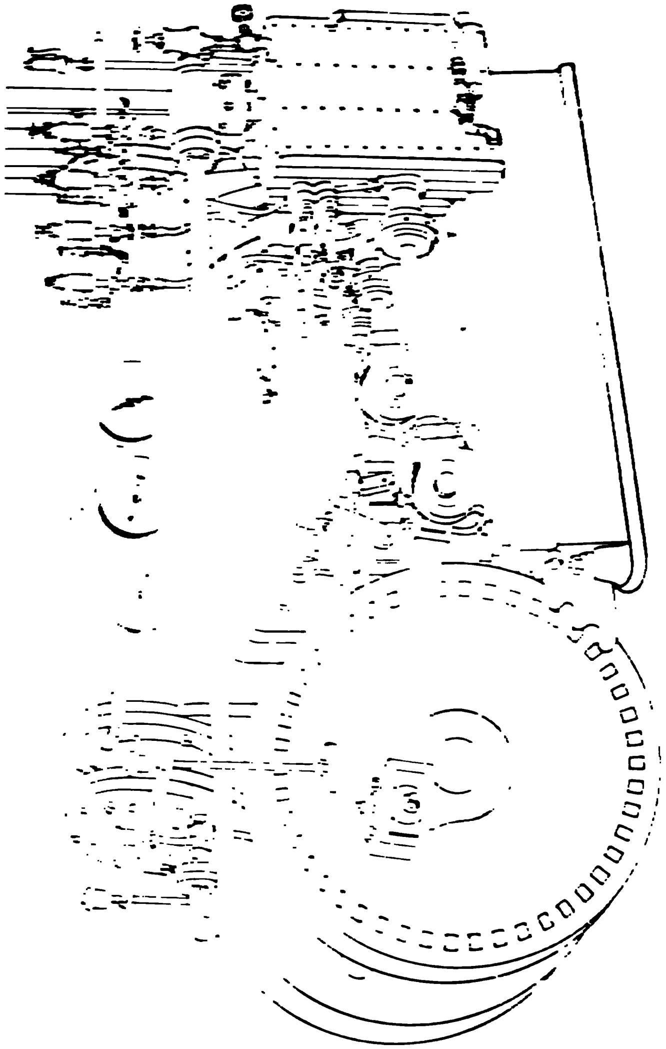


FIG. 50

THE ALLEN DENSE AIR ICE MACHINE

[illegible] is discharged into piping which [illegible] is thus carried off and the temperature [illegible] of the surrounding water.

Should the air be liberated at this time there would be but a slight reduction in temperature, probably from 20 to 30 degrees Fahrenheit.

Expansion of Air. From the cooling coils, the air passes to an expanding cylinder which is not unlike a steam engine cylinder. The air at the pressure of 150 pounds expands and drives a piston which is connected to the driving shaft of the machine and aids in the compression of air. The amount of work accomplished by this cylinder is in proportion to the heat the air contains, and this heat is used up or converted into power. The air is delivered into the cooling pipes for ice-making or refrigeration at a temperature of from 40 to 65 degrees below zero, the final condition depending upon the efficiency of the machinery.

The machinery represented by the accompanying drawings and descriptions is in use in the United States Navy and has been adopted as standard.

First. Assembled parts of the machine. This can best be understood by reference to Fig. 50. The machine is known as the Allen Dense Air Ice Machine.

In Fig. 51 we have three cylinders with slide valves, not unlike a steam engine, A in fact representing a steam cylinder, driving the air compressor B. This cylinder is operated by a single D valve and controlled by a throttling governor or by any governor adapted to conditions at sea. Power from the steam cylinder is conducted by connecting rod and disc crank, to which is attached a driving shaft H with a crank in the center of the shaft which drives the air compressor. On the opposite side of the shaft is a disc crank from which the expanding cylinder is attached which in turn aids the steam cylinder in its work. F is a plunger piston pump for circulating sea water around the coils in the cooling tank C. G is a priming pump for air.

In the *location of cylinder cranks*, the crank driving the air compressor leads the steam cylinder about 30 degrees. This is the practice followed by the best engine and air or gas compressor builders, where they are direct connected. The object of the lead is to apply the greatest pressure attainable to the piston of the air compressor at the time it is completing its stroke, when the angle of the crank pin nears the position exerting the greatest effort on the crank and previous to the time of the closing of the cut-off on the steam cylinders. By this method a much lighter fly wheel

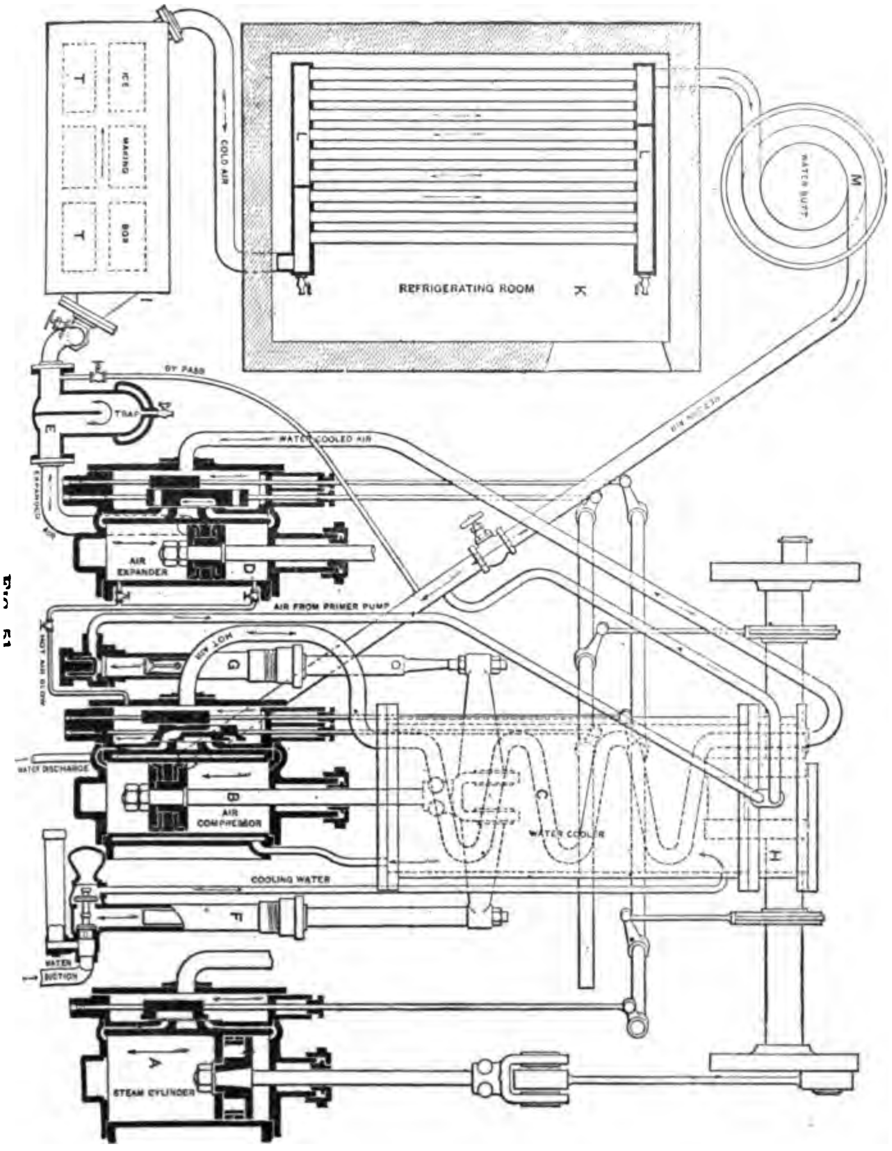

STEAM CYLINDER
A
WATER SUCTION
F
COOLING WATER
AIR COMPRESSOR
B
WATER DISCHARGE
WATER COOLER
C
H
G
HOT AIR
HOT AIR BLOW
AIR FROM PRIMER PUMP
AIR EXPANDER
D
EXPANDED AIR
E
TRAP
BY PASS
WATER COOLED AIR
RETURN AIR
REFRIGERATING ROOM
K
L
L
M
WATER BUTT
COLD AIR
ICE MAKING BOX
T
T
I

Fig. 51

can be used, for the power developed by the engine is applied directly through the shaft and not transmitted to the fly wheel to be given off when the compression of the air cylinder is taking the maximum power and the steam crank is nearly completing its stroke.

Air Compressor "B." In considering the action of this specially designed air compressor, we find that the suction is taken through the opening in the bottom of the valve chest, the opening being located in the same way as the exhaust on a slide valve engine. The discharge is through the face of the valve chest, the location again being similar to the steam supply to a slide-valve engine. The valves are of the slide design. It is a question with engineers whether or not this design is superior to the poppet valve as used by ammonia and kindred gas compressors. The advantages of using the slide valves are: first, they are comparatively noiseless; second, they are not constantly hammering their seats or faces in closing, and third, there is no unbalanced pressure of the valve seat to be overcome by the engine, or to cause the rapid destruction of the metal forming a poppet valve.

Note: For a description of unbalanced valve operation see page 93.

In the operation of the valves we find two rocker shafts, each of which is operated by an eccentric from the main shaft. The rocker shaft nearest to the main shaft operates the valve admitting steam to the engine A; this is a plain D valve. The shaft also operates the rider valve on both air compressor and expander cylinder.

The second rocker shaft operates the main valve of the compressor and the expander cylinder. From the crosshead of the air compressor an arm extending horizontally operates the charging air pump and also the water circulating pump. Water from the water cooler is continuously flowing around the jacket of the air compressing cylinder. The action of the expanding chamber is the same as though steam were used to move the piston, the energy given by this expansion of air aids in driving the air compressor, thereby adding to the economy of the operation. This is not, however, the primary cause for this construction. By using the air, which is at high pressure, to obtain power, the heat that it contains is consumed or converted into power and at 60 pounds per square inch, which is the back pressure carried, there is ob-

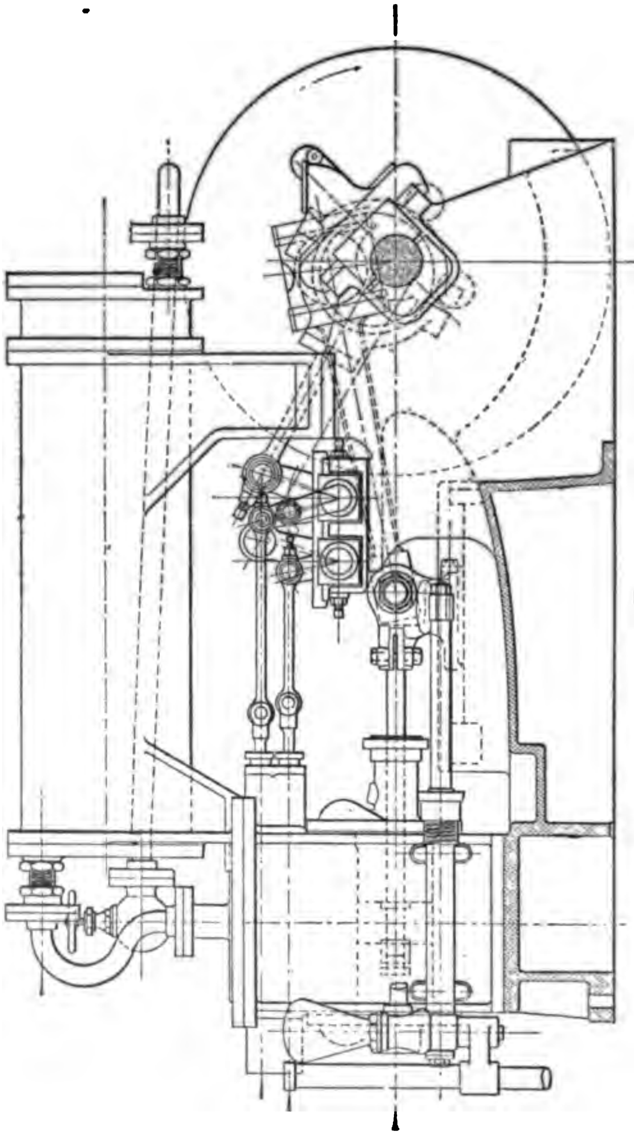


FIG. 52

tained a temperature of from 70 to 90 degrees below zero. This can be obtained with a working initial pressure of 260 pounds per square inch. In no instance is the latent heat of the air affected except to a slight degree, equaling in ratio the ability of the air to carry latent heat due to the different specific temperatures during the compression and expansion.

Relief of Valves. Should the pressure in the expanding cylinder become greater than the pressure in the discharge chamber, due to the distortion of rods or slipping of eccentrics, the valves would spring back from their seats and relieve the pressure. Upon the receding of the piston the valve would close.

Unbalanced Valve Pressure. In considering this question, let us take a poppet valve for our illustration; this is shown in Fig. 53.

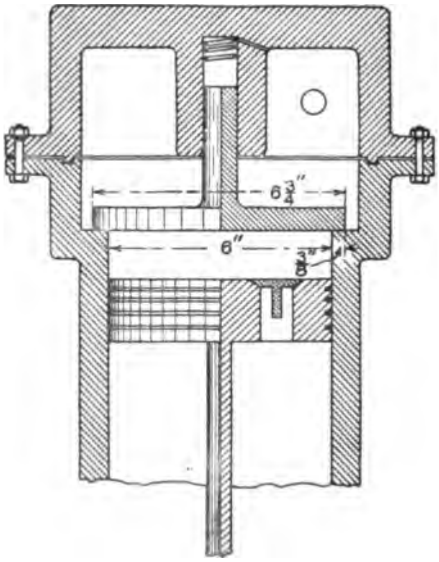


FIG. 53

The design calls for a 6-inch compressing cylinder with a valve seat of $\frac{3}{8}$ inch face giving a bearing surface on top of the valve $6\frac{3}{4}$ inches in diameter. The area of the cylinder equals 28.274 square inches, while the area of the top of the valves including the seat is 33.183 square inches. Let us consider a working pressure of 150 pounds per square inch. To open this valve there must be exerted a pressure of $150 \times 33.183 = 4977.45$ pounds. The power exerted on the under side of the valve, when the pressures are equal to that of the discharge chamber, is $28.274 \times 150 = 4241.1$ pounds, or in order to open the valve there must be added $4977.45 - 4241.1 = 636.35$ pounds. In other words, to force the valve from its seat there must be exerted 28.53 pounds per square inch in excess, or a total pressure above the valve of $150 + 28.53 = 178.53$ pounds per square inch on the entire compressing piston. This power is overcome by placing springs or com-

[illegible] the latter [illegible] the most [illegible]

The [illegible] of this effect in [illegible] of power which equals [illegible] the [illegible] requirements for compression of gases.

Oil and Water Traps. In the main discharge line from the [illegible] is placed a separator [illegible] water or oil that may be in the system. [illegible] any time when fresh air is taken [illegible] operation. These [illegible] starting up a refrigerating plant. The [illegible] can be blown off at any time but [illegible] of the plant.

The [illegible] of the air [illegible] any water or oil that may [illegible] in the system. This [illegible] taking place in the [illegible] air to work with. The [illegible] steam when it is [illegible] water, the latter collecting and remaining [illegible]. These traps hang vertically so that the [illegible] the water and oil is one of gravity. In the [illegible] shown as being horizontal. This is [illegible] the arrangement of [illegible]

Ice Making Tank. [illegible] the expanding [illegible] pipes [illegible] many times [illegible]. It is not necessary [illegible] liquid is to [illegible] and [illegible] it back [illegible] in the expanding [illegible]

[illegible] tank is a [illegible] of brine [illegible]. The [illegible] used are formed [illegible] the same as those used for ice making [illegible]. The [illegible] of the air in passing from the ice-making tank [illegible] by the quantity of ice withdrawn. This [illegible] to raise the temperature of the air above zero Fahr. This [illegible] the amount of heat taken from the tank from [illegible] degrees below zero to zero, or about two British thermal units for each cubic foot of air passing through the cycle.

Refrigerator Box. The air from the ice-making tank passes into the refrigerator box coils, entering at a temperature of zero. These coils are made with a manifold at each end with divisions in the manifolds to direct the flow of air, causing a circulation back and forth of three times the length of the box. By this method of distributing the circulation, a small pipe system can be used for the piping, while the headers are of an area equaling the discharge from the expanding cylinder. The required area for the circulating pipes is made up of several of the smaller pipes. This is not a vital or even a necessary point. A large pipe could be erected, giving the entire area of the exhaust cylinder opening. This would increase the friction of the passage of air over the design as illustrated in the plan.

Water "Butt" or Cooler. From the refrigerator box, the air passes through a coil of pipe in a tank filled with water. The object being the cooling of the water for the use of the men. This is the last operation of the air and it is desired to have the air at as high a temperature as is consistent with the work.

The application of compressed air was first made successful in ocean steamships carrying meat to Europe. The engines were so large that no auxiliary boiler could furnish sufficient steam; so that to cool the refrigerator box it required the starting of one of the ship's main boilers. The air was exhausted from the box at atmospheric pressure, and from the expanding cylinder it was discharged directly into the refrigerator box. The result was that the oil used for lubrication became very strong in the air and left a taste in the meat. The moisture in the atmosphere was turned to snow as it came in contact with the cool air; this snow in turn becoming melted when striking the sides or floor of the refrigerator or coming in contact with the merchandise in shipment. This has been remedied by enclosing the air in piping and having a continuous cycle of the cooling fluid in operation.

In no instance does the liquefaction of air enter into the problem of refrigerating with air.

In the work of construction, iron pipe may be used, with any of the pipe cements for joints common to first class piping.

The following instructions for starting and operating the Allen Dense Air Ice Machines are used by the United States Navy.

On starting the machine, have the blow valves of the expander and the pet-cocks of the various traps open until no more grease or water discharges.

The two 1½-inch or 2-inch valves of the main pipes must be open and the 1-inch by-pass pipe closed; also the ½-inch hot-air valves from the compressor to the expander cylinder must be closed.

Be sure that the circulating water is in motion.

The full pressure is 60 to 65 lbs. low pressure, and 210 to 225 lbs. high pressure.

During the running, open the pet-cocks of the water-trap which takes the water out of the air from the primer pump, frequently enough that it will never be more than half filled. If the water should be allowed to enter the main pipes, it is liable to freeze and clog at the valves.

By keeping all stuffing boxes well lubricated by the lubricator cups, the pressures are easily maintained with but little screwing up of the packing.

If the low-air pressure is not maintained, the fault is almost always due to leaks at the stuffing boxes. Under all circumstances it is due to some leak into the atmosphere, as we have never yet found the primer pump valves at fault.

The packing of valve stems and piston rods consists of a few inner rings of Katzenstein's soft metal packing, then a hollow greasing ring, then soft fibrous packing (Garlock packing).

The sight feed lubricators of the compressor and expander should only use a light pure mineral machine oil, from which the paraffine has been removed by freezing—usually three drops per minute in the compressor and one or two in the expander.

The pistons of the compressor and expander cylinders are packed with cup leathers, which commonly last about one or two months of steady work. When these leathers give out, the high pressure decreases in relation to the low pressure, and the apparatus shows a loss of cold. A leak at any other point of high pressure into low pressure will have the same effect.

These packing leathers are made of thick kip leather, or of white oak-tanned leather of somewhat less than ⅛-inch thickness. They are cut ⅝ inch larger in diameter than the cylinders. The leathers must be kept soaked with castor oil and must be well soaked in that before using; and a tin box containing spare leathers and castor oil must be kept on hand.

Once, or sometimes twice, a day, it is necessary to clean the ıchine by heating it up and blowing out all the oil and ice posits. This is done as follows:

The 1-inch valve of the by-pass is opened. Then the two 1½ 2-inch valves in the main pipes are closed; then the two ½-inch lves in the hot air pipe from the compressor chest to the expander ɜ opened, and the 1¼-inch valve of expander inlet is closed rtly; then the live steam is let into the jacket of the oil-trap wly, keeping the outlet from the steam jacket open enough to ain the condensed steam.

Run in this manner for about one-half hour, during this time ɜquently blow out the bottom valve of the oil-trap, also the blow- from the expander, until everything appears clean. Then ıt off the steam and drain connections of the jacket of the trap d the hot air pipe from the compressor to the expander. Then en the two 1½-inch valves in main pipes. Then close the 1-inch -pass pipe and all pet-cocks and run as usual.

Whenever opportunity offers to blow out the manifolds of ɜ meat-room and the ice-making box (that is, whenever they ɜ thawed), this should be done.

If it is suspected that a considerable quantity of oil and water ve got into the pipe system and are clogging the areas and coat- ʒ the surfaces, the pipes can be cleaned by running hot air rough them as is done during the daily cleaning of the machine. ıe oil and water are then drawn off at the bottom of the ice- ıking box and the manifolds of refrigerating coils.

The clearance of the two air pistons and of the primer plunger only ⅛ inch; therefore not much change of piston rods and con- cting rods is permissible, and when the piston nuts are un- ewed to change the piston leathers, the rod should be watched ıt it does not unscrew from the crosshead.

Whenever it is noticed that the brine freezes, more chloride calcium should be added and should be well stirred into the ine.

ABSORPTION SYSTEM.

By the rapid vaporization of water in the desert of Arabia, e to the intensely dry atmosphere, the first ice, other than that rmed by the natural process of winter freezing, was made. For ɔling large quantities of water, this process is in use to-day on more clearly defined and greatly enlarged plan known as the

Water Cooling Tower. This method is being extensively used for cooling for condensing ammonia gas or for condensing the exhaust steam from an engine.

Carré's First Ice Plant. From the use of this system of vaporization, Carré, a French scientist, constructed the first device for cooling by mechanical means and producing ice, and all devices for absorption refrigeration owe their results to the work so ably accomplished by him.

A definition of the terms used in the two systems in connection with refrigerating machinery is as follows:

Absorption Refrigeration. In the absorption, as in the compression system, it is necessary to maintain pressure upon the ammonia gas when freed from water in order that it may become liquefied, and during its period of liquefaction give up the latent heat.

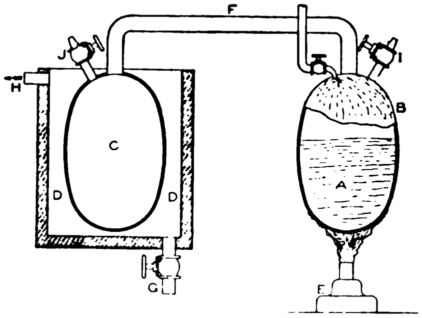


Fig. 54.

The Carré original machine consisted of two strong iron jars connected together with an iron pipe as shown in Fig. 54. In the jar A, or ammonia still, was placed a quantity of highly charged aqua ammonia. A spirit lamp E was placed under the jar. An air cock was placed at I, and when the jar became heated, the air cock was opened until the air was exhausted. The pressure increased in both jars and connection A, B, C. Jar C was placed in a tank and surrounded by cold water D. The supply of this water was constantly changing so that at all times the water remained cold or at its original temperature, which we may assume at 60 degrees Fahr. The heat from the lamp caused the ammonia to vaporize and a pressure was obtained sufficient to liquefy the gas in the jar C. This pressure was about 120 pounds per square inch. The intense pressure made by the ammonia gas was sufficient to raise the boiling pressure of the water to 230 degrees Fahr., thereby preventing the vaporization of the water. This temperature was ample to drive out a large amount of gas.

After the process was complete, which was determined by the length of time that the heat was exposed to jar A, the lamp was removed. The water was drawn off from around jar C and the tank filled with whatever substance it was desired to freeze. A water pipe allowed water to flow over the jar A and cool its contents, the result being that the pressure was removed from both jars A and C, and the liquid anhydrous gas began to vaporize in jar C, the gas returned to jar A, and was taken up in the water from which it had been expelled. By the passage of the anhydrous ammonia from a liquid into a gas, the heat which was taken up and caused the vaporization, being held latent by the gas, was taken from the surrounding liquid in tank D.

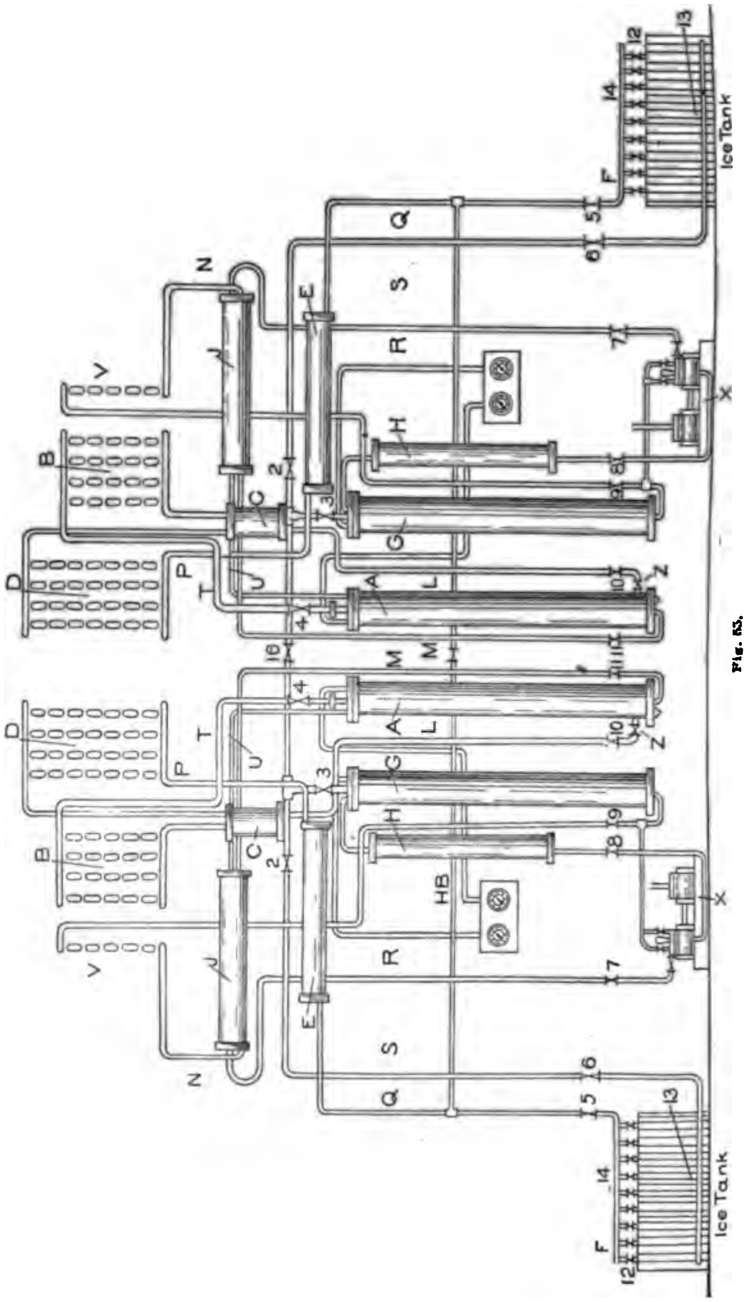
This produced a low temperature, the boiling point of liquid anhydrous ammonia being at atmospheric pressure 47 degrees below zero Fahr. The ratio of the temperature and pressure determines the temperature at which the gas boils in passing from a liquid to a gaseous condition.

Reabsorbing. As the expanded gases return to the jar A and come in contact with the water, they are absorbed by the water and in this process of absorption, the heat, which is taken up latent by vaporization in the freezing jar C, is given off. The heat thus given off in the jar A is carried away by the water flowing from connection H.

It was by this simple device that Carré established the principles of refrigeration. We should remember that it was in 1858 when this work was done and that the methods of working iron were then very crude. Carré's success was due to his patience and endurance.

In Fig. 55 is shown a double plant absorption machine, including an ice making tank, each machine having a capacity of 12½ tons, or a 25-ton ice-making output for every twenty-four hours of constant operation. The plants are connected together so that they can be interchangeable in their working parts.

The following is a brief description: An ammonia still A is built of iron pipe 12 inches in diameter with heavy flanged heads, which are of wrought iron, screwed on to the pipe and soldered where the pipe and metal of the head joins. The object of a flanged head is to admit of coils in the interior. A strong solution of aqua ammonia is placed in the still and heated by a steam spiral coil placed in the interior of the still. In operation, the aqua ammonia still is kept about one-third filled with aqua ammonia. In naming the parts of the apparatus, each designer or builder selects names to suit his fancy. While the part used for applying heat to the aqua ammonia is necessary to all designs of absorption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F
Ice Tank

Fig. 83.

achinery, we find it bearing the following names: "ammonia iler," "ammonia still," "ammonia pressure generator."

The steam connection is not shown in the drawing. The am- onia gas is driven off by heat and passes up through the connec- n T into the dehydrating coil B. As the ammonia gas passes om the still, it carries with it a small quantity of water in a foam- g condition which is separated from the gas in the dehydrating il. This coil is enclosed in a tank and filled with water which is aintained at a temperature of approximately 150 degrees Fahr. this temperature the water condenses, but allows the ammonia remain a gas. The pressure in the still corresponds to the tem- rature being maintained in the condenser for liquefaction. Here e same ratio is obtained as in the compression machines, usually om 110 to 180 pounds per square inch. It is usually considered cessary to maintain 150 pounds per square inch for liquefaction. is requires condensing water at approximately 60 degrees.

From the dehydrating coil the gas and water pass to a sepa- e tank C, which is a plain cylinder. The action is to separate e gas from the water, the gas passing from the top of the sepa- ing tank C to the condenser coil D, while the water which has en separated from the gas by the action of the dehydrating coil ops by gravity through pipe L and valves 10 back to the still.

The law of gravity holds good in all cases and under all cumstances where pressures are the same. Therefore, in all rts of the ammonia generating plant this law of falling of liquids gravity is made use of and care must be taken in designing nts that liquids will come to their proper places by gravity.

The ammonia gas is now separated from water and passes the condenser for liquefying.

Ammonia Condensers. The condenser coils D may be en- osed in a tank and surrounded with water, or they may be left en to the atmosphere and the water allowed to drip over them. the ammonia condenser, the heat held latent by the ammonia s is given off and a liquefaction takes place, the liquid ammonia ving the appearance of clear water. The ammonia drops into iquid storage tank E, through connection P.

Freezing or Ice Making Tank. The ammonia is now uefied and ready for use under a pressure normally of 150 unds per square inch.

Note: Do not confuse liquid ammonia with anhydrous ammonia.

From the liquid tank E, the liquid ammonia flows through the connecting pipe Q, being held in check by the main liquid valve 5. The object of this main liquid valve is to simplify the working of the plant. The nine valves 12, are the expanding valves, each connected to the coil of pipe in the freezing tank. Each coil has a separate return with a stop valve, entering connection 13. The tank is shown with insulation but the coils are not shown. This freezing tank is designed to hold 240 cans or ice moulds, each having the capacity of 200 pounds of ice, when the freezing is complete. The tank is eight cans wide and thirty cans long, or a tank 9 feet wide by 60 feet long and 36 inches deep, inside measure. These tanks may be built either of iron or wood as the designer wishes. The coils run the entire length of the tank, between the cans and outside of them.

Freezing. In operating, the main liquid valve 5 is opened and the gas flows to the nine expanding valves 12, which are opened to permit a flow of gas through an opening varying from $\frac{1}{100}$ to $\frac{1}{8}$ of a square inch.

Low Pressure Side of the Apparatus. At this point the gas is allowed to pass from the pressure under which it is condensed to the vaporizing pressure, usually from 150 to 15 pounds per square inch.

Note: In all cases where pressures are referred to, the zero point is the atmosphere and not absolute zero.

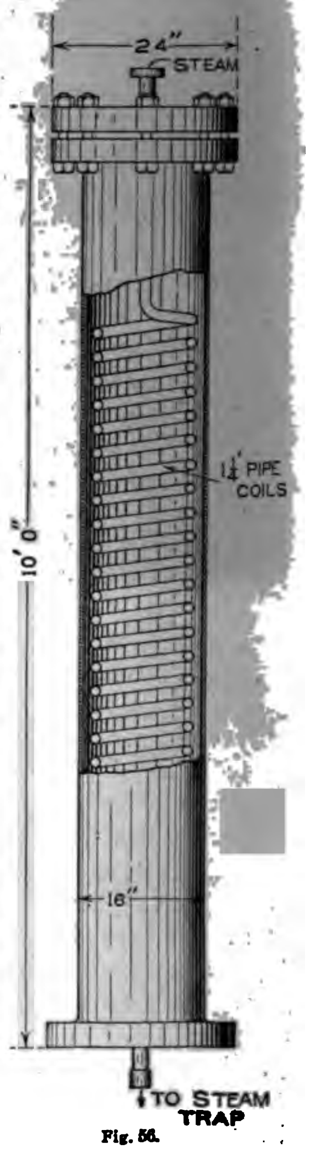
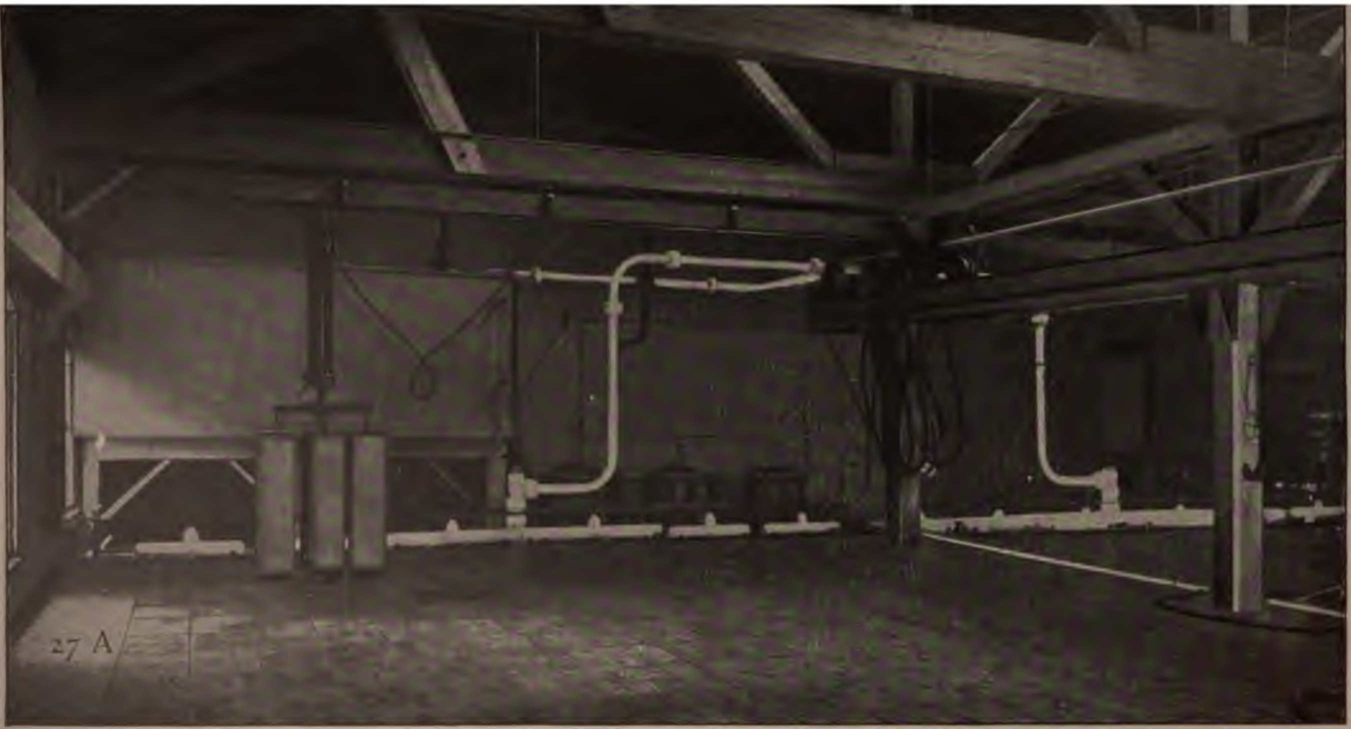


Fig. 56.

The liquefied ammonia entering the freezing coils under pressure of 15 pounds per square inch immediately begins to vaporize and in so



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;akes the specific heat from the water. The heat passes ı the iron pipes, is taken up and returned to the absorber G ı connection S. The boiling point of liquid ammonia at 15 pressure approximates 26 degrees below zero Fahr. The of the ammonia in boiling is identical to water boiling at grees in the atmosphere, with which every one is familiar.

ɛsorber. The vaporized gas flows into the top of the ab- where it meets the water from which the gas has been

rcuit of Weak Water. Returning now to the ammonia uring the period of distillation, the water, freed from the ıeat and called by different names such as "mother liquid," water," etc., is allowed to flow from the bottom of the still ı connection, M, and passing through "heat exchanger" or zer" J, flows on to weak water cooler V, entering the cooling :he bottom. It is preferable to have this coil enclosed in a as to get the advantage of "Baudlett" system of cooling or ıging of the hot water to the hot gas and cold water to the s. From this cooler, the weak liquid returns to the ab- 3, through valve 9. The water is carried to the top where down over a spiral coil of water pipes. The water and ɜ intermingle and the water absorbing the gas is allowed to o reservoir H. A strong solution of aqua ammonia is taken ɔump through connection 8 and returned to the still through ion 7, there to be re-distilled and the cycle of operation ɔ be performed.

essure Gages. There are two pressure gages: one being a ɜssure and the other a low-pressure instrument. The high- ɜ gage reads from zero to 300 pounds or more, the low- ɜ being a combination gage reading from 30 inches vacuum pounds pressure per square inch, the zero point being at ıere and not absolute. These gages are connected as fol- he high pressure gage to the head of the ammonia still A, pressure gage to the head of the absorbing tank G. Stop ıust be placed in the gage lines between absorber and still- ads to permit the removal of the gages for cleaning or In locating gages, a position about 12 inches higher than ı should be chosen. At this point they are easily seen and le for cleaning.

A steam gage is required on the steam line entering the still, also a reducing valve on the steam line to control the pressure of steam. The steam gage is connected between the reducing valve and the ammonia still steam coil.

A steam trap is placed on the discharge of the steam coil coming from the still, and so connected as to keep the still coils free from water at all times.

Glass Gages. It is common with many builders of refrigerating and ice-making plants to place on their systems glass gages, in order to determine the amount of liquid in the different parts by actual observation. Many accidents of a severe nature have been the result of breaking these glass tubes.

Automatic Gage Cocks. There have been many devices to close the gage cocks automatically in case of breaking the glass; one device being the placing of long lever handles on the plug cocks and closing them by a rope over a pulley at a safe distance. These glasses are often made 36 inches in length, and from that down to a few inches.

Enclosed Glass Gages. It is often the practice to enclose these glass tubes within a pipe split for a quarter of an inch on opposite sides and the space between the glass and the tube filled with plaster of paris; or, to make the tubes quite thin and drill holes through them, the drill passing straight from side to side and at right angles to the pipe, the object being that when the glass bursts the pipe covering keeps the glass from flying and injuring the attendant.

Glass Gages in Operation. It is a fact that most gages after they have been in use about six months become filled with a dirty substance, and become inoperative, their reading being misleading to the attendant. To the successful operation of any refrigerating plant, either compression or absorption, the presence of glass gages is an unnecessary adjunct, and the attendant soon becomes so familiar with his plant that he can operate it without reference to glass gages.

In installing and first charging a plant, these gages are a convenience but not a necessity. In order to charge a plant without gages, the engineer must know positively the amount of liquid necessary to fill the different parts of his apparatus. Often thermometers are inserted to determine the temperatures of the liquids at the different parts. This may be very important to determine the positive condition where a test is made for laboratory purposes.

In the parts of the apparatus shown in Fig. 55, glass gages could be located on ammonia still A about 30 inches from the bottom of the tank; a glass about 24 inches long should be used. The water collecting at this point should fill the tank to a height of about 45 inches from the bottom, which would bring it to the center of the glass. A thermometer could also be inserted in the still 6 inches from the bottom, this to be an angle thermometer and a "thimble" inserted to come in contact with the liquid. In this thimble the bulb of the thermometer should be placed. This thermometer should read from 80 to 300 degrees Fahrenheit.

Absorbing Tank Glass Gages and Thermometer. Glass gages and thermometer may be placed on this tank in the same way as those described for the distilling tank A. On liquid receiving tank E glass gages may be placed, the connection being made in the head or from the upper and lower sides of the tank. This glass is to indicate the quantity of liquefied ammonia in the tank.

Operating Without Glass Gages. The object of a refrigerating or ice-making plant is to do the work for which it is designed, and when an absorption plant is operating and the output is not equal to the amount that it has turned out when operating at its best, either it is short in some requirements necessary to make it work successfully, or there is present a condition that is foreign to the best working condition. No glass gages can determine these conditions for the attendant. Only familiarity with his plant will instruct him what to do. Passing his hand on the weak water pipe as it leaves the still and also on the return for the water entering the absorber, will tell him the conditions necessary for successful operation.

Gas Foreign to Ammonia. One of the most serious obstacles to the successful operation of absorption ice-making machinery is the generating with the ammonia still of a superabundance of free hydrogen gas, ammonia gas being composed of three parts nitrogen gas.

There is present in the still, water which forms the method of transit of the ammonia for part of its journey. This water is formed of two parts of hydrogen to one part of oxygen. The iron

ming the heating coils has a large capacity for taking
d holding it in the shape of iron scales or rust. The
he ammonia tends to free this rust or oxygen scale from
on which it has formed and as a result a large deposit of
s in the ammonia still. The worst feature is that the
of the oxygen and iron has liberated a large amount of
gen gas which finally rises to the highest point in the
ually in the condensers, and remains there where the
are not right for it to liquefy, because there is not
itrogen gas to combine with it and form new ammonia.
It is necessary to get this hydrogen gas out of the system. This is done by having a valve at a convenient place and occasionally blowing it off. In a refrigerating plant in operation in 1904, which was built to do refrigeration equal to the melting of 12 tons of ice for every twenty-four hours of operation, the actual result upon a careful observation, where the conditions permitted a knowledge of the facts, was 2.6 tons. Part of the defect probably might have been due to a depreciated charge of ammonia gas.

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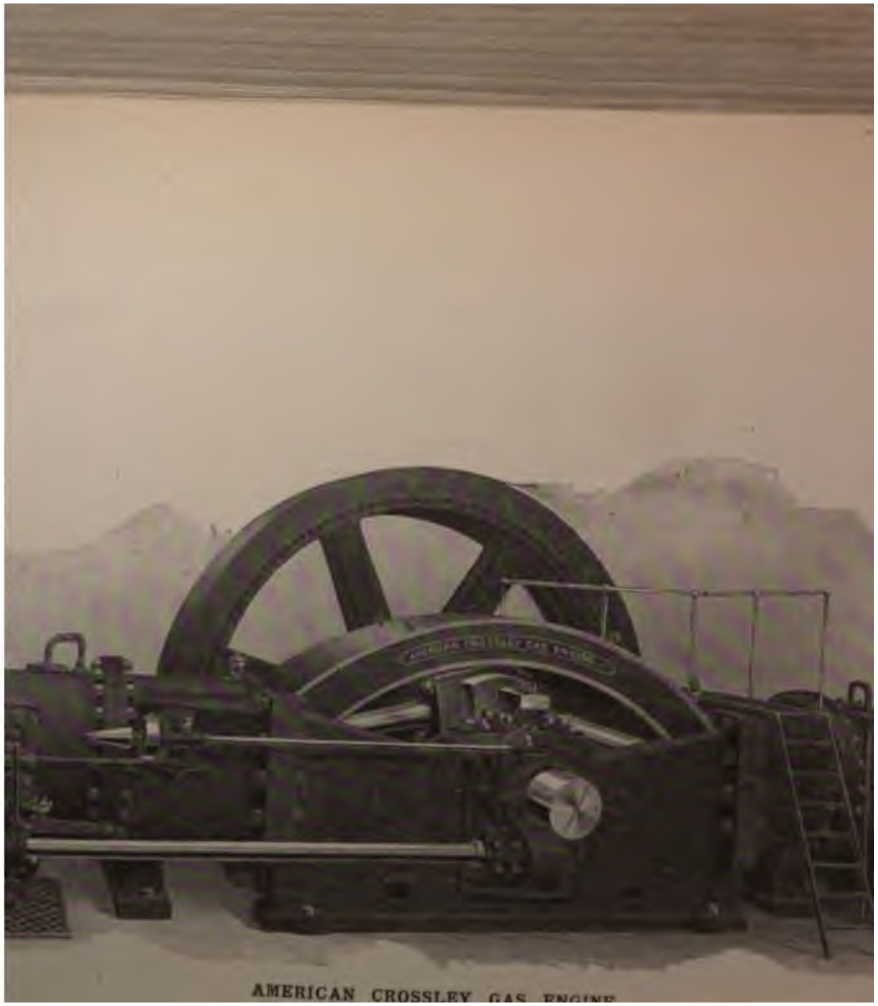


AMERICAN CROSSLEY GAS ENGINE

GAS AND OIL ENGINES.

The heat engine is at present the most important of all the available generators of power. Its purpose is to convert the heat of combustion of some fuel into work. Existing heat engines may be divided into two classes, according to where the combustion of the fuel takes place. In one class the combustion takes place entirely outside the working cylinder, and the heat of combustion is transmitted by conduction through the walls of a containing vessel to the substance which does the work. Such engines may be called *external combustion motors.* The most common example of this class is the steam engine; another example, which is but little used, is the hot-air engine. If the combustion takes place inside the engine itself, or in the communicating vessel, so that the products of combustion act directly on the engine, we have an engine of the second class—the so-called *internal combustion motors.* Gas and oil engines are the common examples of this type of motor.

THE EXTERNAL COMBUSTION MOTOR.

Engines of the second class have certain inherent advantages over external combustion motors. In the steam engine, practically the most perfect of the external combustion motors, the heat of combustion which is generated in the furnace passes through the plates of the boiler to the water on the other side. During this process about twenty-five per cent of the heat is wasted by radiation and by loss up the chimney in good modern plants. The water in the boiler is heated to a temperature which does not exceed 400° F. because at that temperature it has a pressure of nearly 250 lb. per square inch. If the water were heated to a much higher temperature, the pressure would be too great (for example, at 500° F. the pressure would be 700 lb.), requiring boilers and engines stronger than are at present practicable. The products of combustion in the furnace have a temperature which is seldom less than 2,000° F. There is consequently a very large necessary drop of temperature as the heat passes through the boiler plates. The proportion of the total heat going to an engine that can be converted by the engine into work depends chiefly upon the temperature range of the working substance, and in the steam engine this range is made comparatively small—not exceeding 300° F.—because of the corresponding pressure limits. Consequently a steam plant not only loses much of its heat up the chimney, but also is able to convert but a small part of the heat that goes to the engine into work. In the best modern steam engines about twenty per cent. of the heat going to the engine is converted into work, and about sixteen per cent. of the heat of combustion of the fuel is converted into work in the best modern steam plants. The ordinary steam engine does not convert into work more than from six to ten per cent. of the heat of combustion of the fuel. An economical steam plant consists not only of boilers and engines, but also has a large number of auxiliaries, such as feed pumps, air pumps, condensers, feed water heaters, economizer, coal conveyers and steam traps. It requires considerable time and fuel to raise steam in the boilers before the plant can be put in operation after shutting down; or, if the fires are kept banked, so as to keep up steam pressure while the engines are not running, a not inconsiderable amount of fuel will be used for this purpose, without any corresponding work being done.

THE INTERNAL COMBUSTION MOTOR.

In the internal combustion motor where the fuel is a gas or volatile oil, there is no apparatus corresponding to a boiler and no losses corresponding to the boiler losses. If the fuel is coal, it has to be converted into gas before it can be used in an internal combustion motor, and thus necessitates the use of a gas producer in which some heat will be lost, though not so much as is usual in a boiler. The fuel being burned in the engine gives there a temperature of from 2,000° F. to 3,000° F., so that the temperature range in the engine is very large, and consequently the engine can be more efficient—that is, can convert a larger proportion of the heat of combustion into work than in a steam plant. The high temperatures are not necessarily accompanied by high pressures because it is air and not water which is heated to that temperature. In practice the best internal combustion motors have converted thirty-five per cent. of the heat of combustion into work, or twice as much as the best steam engines, and the ordinary small gas engine will convert from fifteen to twenty per cent. of the heat of combustion into work. The internal combustion plant is also much simpler, having but few auxiliaries. The number of men necessary to run a large gas engine plant is small, the plant is ready to start up at a minute's notice, and the standing losses are very small or nothing. When a liquid fuel is used, the absence of boiler or other auxiliaries makes the internal combustion motor lighter, more compact, and more easily portable than any other motor. The absence of a boiler also does away with the risk of a disastrous explosion, and consequently there is no inspection required by law, no license for running the plant, and lower rates for insurance. The internal combustion motor is comparatively recent in its practical use. The last five years have brought about great improvements in its operation, a big increase in its use and large extension in its application. The internal combustion motor is less uniform in its speed of rotation, and is more liable to derangements than the steam engine, but these difficulties are rapidly being overcome, so that modern gas engines are used for electric lighting and have a reliability but little short of that of the steam engine.

The fuels that are used in external combustion motors may be solid, liquid or gaseous. In internal combustion motors, though, theoretically the fuel may be in any of these three forms, the solid fuels are not practicable because the incombustible matter or ash present in them would rapidly destroy the rubbing surfaces in the cylinders. The actual fuels used are either gaseous or liquid, and the latter may be sent into the cylinder either as a vapor or as a liquid. There is no essential difference between engines using gas and those using oil; the cycle of operations occurring in the cylinder is the same with both kinds of fuel; the only differences are slight structural differences, with the addition of special apparatus for vaporizing the oil. The same engine can be, and often is, converted from a gas to an oil engine by merely changing valves.

In this paper whatever there is of thermodynamic theory applies to both gas and oil engines. The special features of oil engines are treated after the discussion of the gas engine.

HISTORY OF THE INTERNAL COMBUSTION MOTOR.

The history of the internal combustion motor begins with the invention of cannon. A gun is a motor in which the working substance is the gas resulting from the combustion of the powder and in which work is done on the projectile, giving it kinetic energy. Such a motor is not continuous in its action, but it offers possibilities of a practicable engine if the powder charge is small and the projectile or piston on which the gases act is restricted in its movement. The earliest internal combustion motors devised for doing useful work were intended to use gunpowder. The first of these was suggested by Abbé Hautefeuille in 1678, and was followed shortly by others, none of which were practically realizable in the then state of the mechanic arts.

It was not till the discovery by Murdock, near the end of the eighteenth century, that a combustible gas could be obtained from coal by a process of distillation, that a practical internal combustion motor was possible. As soon as the properties and method of manufacture of coal gas became known, numerous attempts were made to use it in engines. Until the year 1860 many engines were devised, patented, and in several cases constructed, operated and sold. None of these engines can be said to have been satisfactory. They were irregular in action, noisy, wasteful of fuel, and generally had practical defects.

The Lenoir Engine, which appeared in 1860, was the first really practical gas engine. Hundreds of these engines were made and sold, and the greatest interest in this type was aroused in France, where it was built, and in England, where it was largely used.

In general appearance the engine resembles a double-acting horizontal steam engine. The cylinder, shown in horizontal section in Fig. 1, has a separate admission port *a* and exhaust port *b* at each end. The valves are simple slide valves driven by eccentrics, and so designed that the inside edges alone uncover the ports. The valve G is used for the admission of the explosive mixture, which consists of air entering the valve cavity from *d* and gas coming through one of the branches *r* of the gas pipe and passing through the hole *i* in the valve. The air and gas enter the port *a* through a number of small holes in which they are thoroughly mixed, and the mixture is exploded in the cylinder, when desired, by an electric igniter *n*. The exhaust is through the port *b* and the cavity in the exhaust valve H to the atmosphere. As the cylinder rapidly becomes very hot, it is provided with a water jacket.

The series or cycle of operations which takes place in this engine is as follows: During the first part of the stroke the admission valve G uncovers the port *a* so that a mixture of air and gas enters the cylinder, filling the space behind the piston. At half stroke the valve closes the port and a spark from an induction coil passes between the terminals *n* of the electric igniter, exploding the mixture and raising its pressure to 60 or 70 lb. per square inch. The piston is then forced to the end of its stroke, the products of combustion expanding behind it. At the end of the stroke the valve H uncovers the exhaust port and keeps it open throughout the whole of the return stroke so that all the products of combustion are expelled to the atmosphere. A similar cycle of operations occurs on the other side of the piston. In Fig. 1 the valve G is just opening the port at the left so that admission may

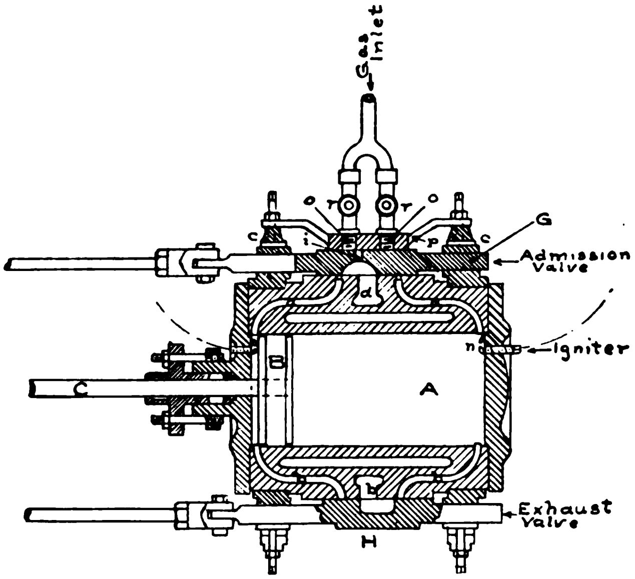


Fig. 1. Longitudinal Section of Cylinder of Lenoir Engine.

take place there, and the valve H is just opening the port at the right so that exhaust may occur from the other end of the cylinder. A reproduction of an indicator card from this engine is shown in Fig. 2.

This engine gave considerable trouble in many cases, but the principal reason for the falling off in its use was the large amount of gas it required. It used from 60 to 70 cubic feet of coal-gas per I. H. P. per hour, or from three to four times as much as a modern gas engine, so that it did not compare very favorably with the steam engine in its running costs.

The Otto Cycle. In the year 1862 it was pointed out by a French engineer, Beau de Rochas, that in order to get high economy in a gas engine certain conditions of operation were necessary. The most important of these conditions is that the explosive mixture should be compressed to a high pressure before ignition. In order to accomplish this he proposed that the cycle of operations should occupy four strokes or two complete revolutions of the engine, and that the operations should be as follows:

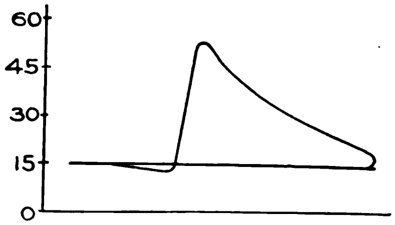


Fig. 2. Indicator Card of Lenoir Engine.

1. *Suction* or *admission* of the charge of gas and air throughout the complete forward stroke.

2. *Compression* of the explosive mixture during the whole of the return stroke so that it finally occupies only the clearance space.

3. *Ignition* of the charge at the end of the second stroke and *expansion* of the exploded mixture throughout the whole of the next forward stroke.

4. *Exhaust* beginning at the end of the forward stroke and continuing throughout the whole of the last return stroke.

This cycle was not actually used till 1876, when Dr. Otto adopted it in his engine and thereby produced the modern gas engine. The four-stroke cycle of Beau de Rochas is now universally known as the *Otto cycle*. In the past twenty years several other cycles, some of great merit, have been devised and used, but at the present day an overwhelming majority of the existing internal combustion motors use the Otto cycle. The engines using this cycle are accordingly treated of first and at greatest length.

THE MODERN GAS ENGINE.

The construction of a modern gas engine using the Otto cycle is illustrated in the sectional elevation, Fig. 3, of a vertical engine. As in practically all Otto cycle engines, the engine is single-acting and has a long trunk piston which acts as a crosshead and also permits the use of several piston rings by which leakage past the piston is prevented even with the high pressure obtained by the explosion. The engine is made single-acting because the cylinder would get too hot for continuous running if it were double-acting; and moreover, a piston rod and stuffing box give great trouble if exposed to the high temperature of the burning gases. Since the cycle occupies two revolutions, the valves and igniter have to operate once in two revolutions, and therefore the cams which drive these parts are mounted on shafts running only half as fast as the main shaft.

Referring to Fig. 3, A is the shaft which carries the exhaust valve cam, and is driven by gears from the main shaft. The exhaust cam works against a roller carried on the free end of the guide lever G. The exhaust valve E has a long stem projecting downward and resting on a hardened steel plate on the upper side of the guide lever G. The spring surrounding the stem serves to bring the exhaust valve back to its seat and to keep the stem in contact with the guide lever. From the exhaust cam shaft A a horizontal shaft with bevel gears leads to the opposite side of the engine, engaging with a vertical shaft which in turn drives the upper cam shaft B. Incidentally, the vertical shaft carries the governor. The upper cam shaft carries two cams. One engages against a roller on the end of the horizontal lever C. As the throw side of this cam comes uppermost, the opposite end of the lever C depresses the stem of the inlet valve J, opening the latter for the admission of the mixture of gas and air. A spring on the stem of the inlet valve furnishes a means for closing it and keeping the cam and roller always in contact with each other. Immediately adjacent to the inlet valve cam is the igniter cam, which at the proper instant operates a horizontal plunger working through the guide D to break the electric current through the wire S at the terminals of the igniter F.

The cylinder heads and the upper end of the cylinder are thoroughly water-jacketed, as, owing to the extreme heat to which these parts are subjected, they would soon become red-hot if no means were provided for keeping the temperature down. The cooling water enters at H and is discharged at K.

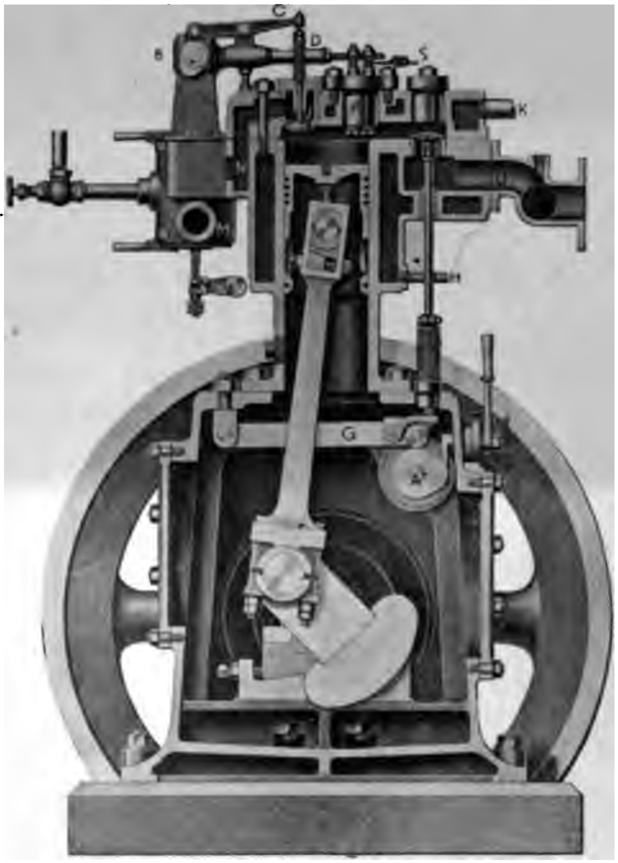


Fig. 3. Westinghouse Gas Engine.

The gas and air enter the mixing chamber M by separate inlets, in proportionate amounts which can be regulated, and the mixture is conducted through a distributing chamber to the port

N leading to the cylinder head in which the inlet valve is located. The exhaust gases escape through O.

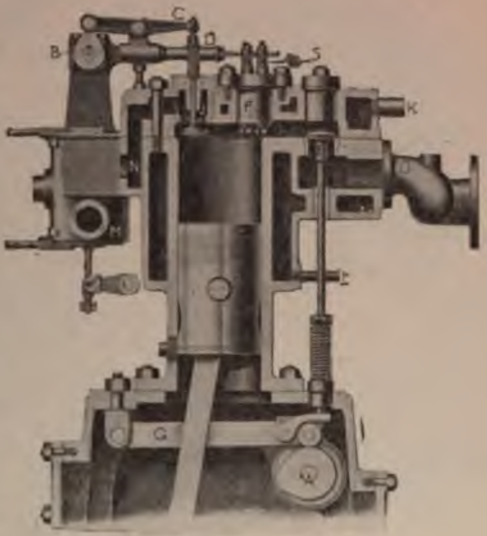


Fig. 4. Admission of the Charge.

The operation of this engine is illustrated in the accompanying illustrations. The admission of the charge of air and gas takes place during the first downward stroke of the engine (Fig. 4). The exhaust valve E is closed and the admission valve J is open, and closes only when the piston is at the end of the stroke and the cylinder is full of the explosive mixture. During the return stroke (Fig. 5) both valves are closed and the charge is compressed till at the end of the stroke it occupies only the clearance space. Shortly before the end of the stroke the igniter cam has brought the igniter terminals into contact, completing an electric circuit. When the crank is nearly on its dead center the igniter terminals are separated by the action of a coiled spring in the guide D, and as they fly apart the circuit is broken and a spark passes between the terminals (Fig. 6), igniting the charge. An immediate

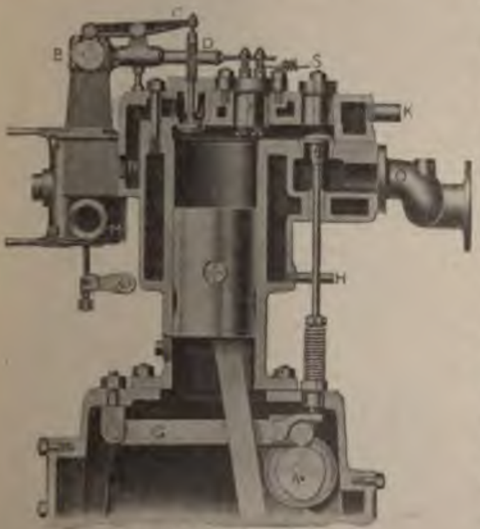


Fig. 5. Compression.

rise of pressure occurs and the piston is forced downward, both valves remaining closed until just before the end of the down stroke, when the exhaust valve E opens.

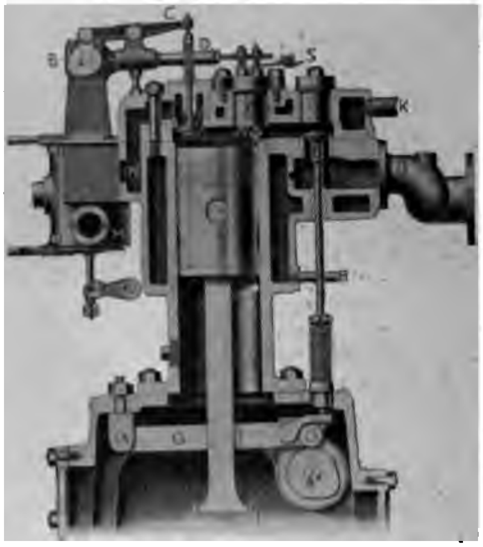


Fig. 6. Ignition.

During the whole of the last return stroke (Fig. 7) the exhaust valve E remains open and the products of combustion are forced through O to the atmosphere. The exhaust valve closes as the piston completes the stroke, and everything is in readiness to recommence the cycle.

Another form of vertical engine using the Otto cycle is shown in vertical section in Fig. 8. In this engine there are three valves—the inlet valve *b* for the gas, the inlet valve *a* for the explosive mixture, and the exhaust valve. All three valves are operated from the shaft *c* which is driven from the main shaft by spur gearing, reducing the speed to one-half that of the main shaft. A cam on the shaft *c* lifts the pivoted lever *d* at the end of which is the long spindle of the valve *a* through which the charge is admitted. The spindle carries an arm *e* which comes in contact with a short link on the

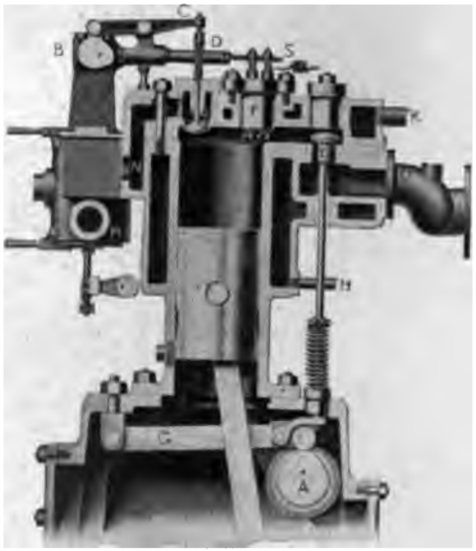
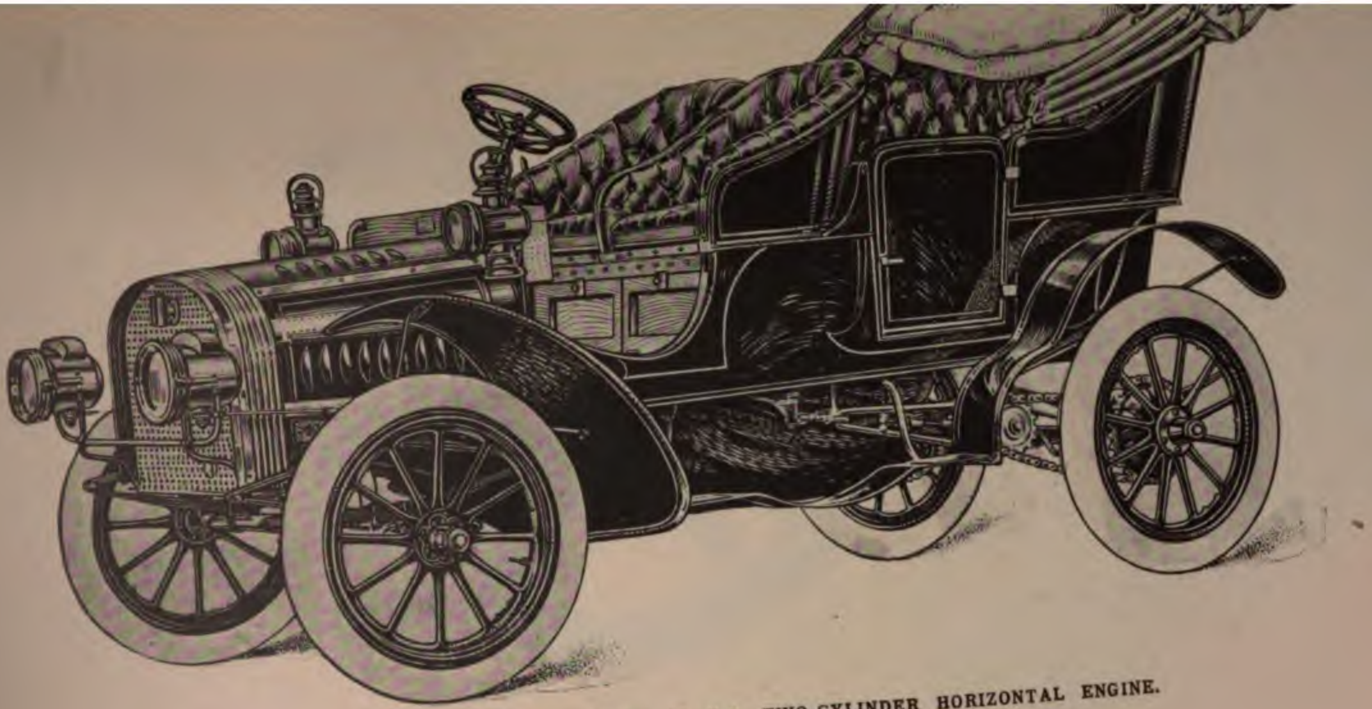


Fig. 7. Exhaust.



SURREY TYPE AUTOMOBILE WITH 20 H.P. TWO-CYLINDER HORIZONTAL ENGINE.
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ASTOR, LENOX
TILDEN FOUNDATIONS

stem of the gas admission valve *b* whenever an explosive charge is required, so that both the valves *a* and *b* are open at the same time. The space *g* behind the valve *a* and around the valve *b* is in free communication with the atmosphere. With *a* open and the piston decending, air is drawn in and thoroughly mixed with the gas while passing through *a*. If the governor throws the short link to one side, the arm *e* does not come in contact with it, the gas valve does not open and air alone is taken into the cylinder during the admission stroke. The exhaust valve is behind the admission valve but is not shown in the diagram.

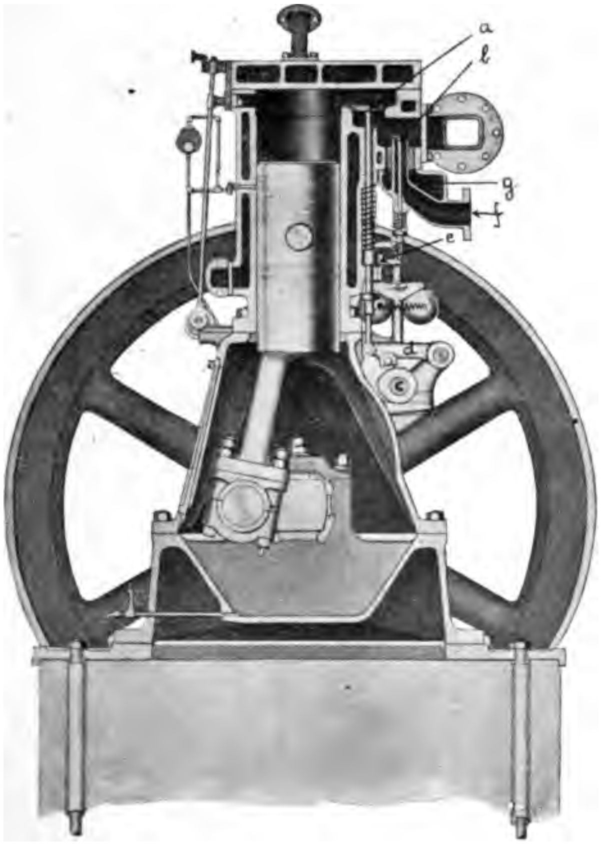

Fig. 8. Nash Gas Engine.

An example of the horizontal form of gas engine is given in Fig. 9, which is a vertical cross section through the cylinder and valves. The valve *a* for the admission of the explosive mixture acts automatically, opening when the pressure in the cylinder falls below the atmospheric pressure. The exhaust valve *b* is opened at the proper time by the action of the cam *c*, which, acting on a roller *d* at the extremity of one arm of the bell crank lever fulcrumed at *e*, pulls the rod *f* to the left and through the bell crank lever fulcrumed at *g* lifts the exhaust valve as shown. The cam *c* is mounted on a shaft which is driven through the spur wheel *k* by the spur wheel *h* (dotted) on the crank shaft. The wheel *k* has twice as many teeth as the wheel *h*, and consequently the cam shaft revolves only half as fast as the main shaft.

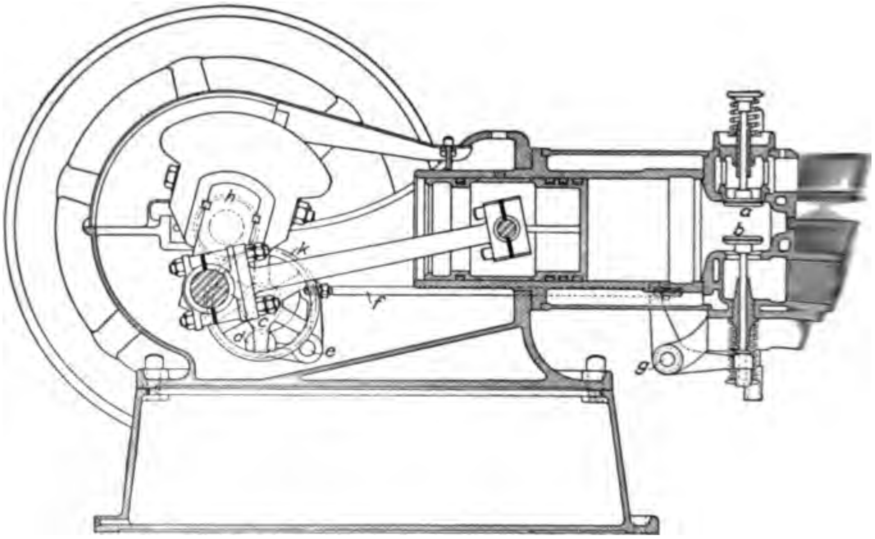

Fig. 9. Olds Engine.

Another horizontal engine is illustrated in Figs. 10, 11 and 12. In this case the admission valve B and the exhaust valve D are both horizontal, a position which can be used satisfactorily only for engines of small size. The air goes to the admission valve through the pipe N, and is shown as taken from the base of the engine. The gas mixes with it at H (Fig. 10), entering throug a nozzle. The amount of gas entering is controlled by the thrott valve A (Fig. 11), and the time at which it enters is determined by the valve G, which is opened at the desired time through the action of the cam P (Fig. 11). The cam P coming in contact with the roller Q at the end of the lever fulcrumed at R, gives a movement to the rod S which is transmitted to the valve through the levers best seen in Fig. 12. The admission valve B is automatic in its action. The exhaust valve D is opened by the action of the cam T (Fig. 10) acting on the roller U at the end of the valve rod W. The valve rod is supported near its free end by the lever X. Both the cams P and T are on a shaft driven from the crank shaft by spur wheels. The governor (Fig. 12) is of the fly-wheel type and consists of the balls which are held by spiral springs and which operate a sleeve on the main shaft. When the engine is above speed, the movement of the sleeve throws the roller Q out of line with the cam P, and consequently there is no admission of gas to the cylinder.

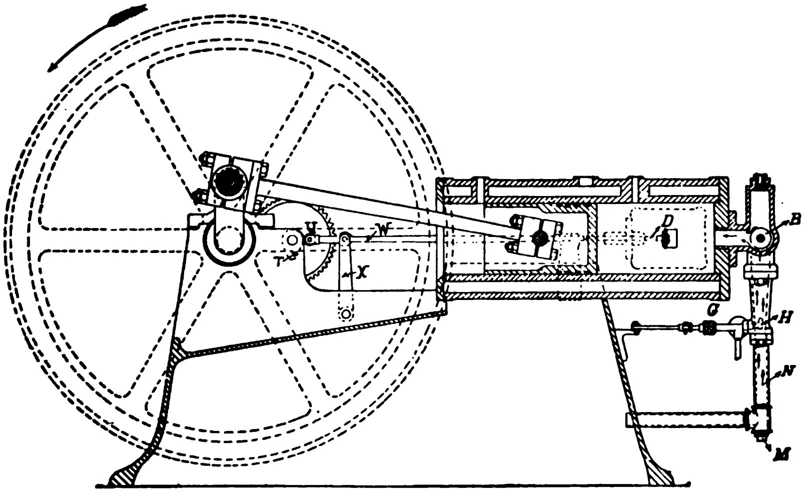

Fig. 10. Charter Engine. Sectional Elevation.

The inlet and exhaust valves in gas engines are nearly always *poppet* or *mushroom* valves similar to those shown in Figs. 4 to 11. The exhaust valves are nearly always mechanically operated; the main inlet valves are often automatic. The automatic valve is similar to a pump suction valve and is kept on its seat by a weak spring (*a*, Fig. 9) and only opens when the pressure in the cylinder is sufficiently less than the atmospheric pressure to permit the latter to overcome the resistance of the spring. Consequently the suction or admission pressure in the gas engine is necessarily low when automatic inlet valves are used. The effect is to decrease the work done by the engine and also its efficiency; the only advantage is the greater simplicity. Most small gas engines have automatic inlet valves.

A positively actuated admission valve is shown in Fig. 13. The valve is lifted by a cam *a* on the side shaft *b* through the lever fulcrumed at *c*. The valve closes by its own weight, assisted by a spring, and is guided in its motion by a long sleeve. The valve chest is completely water-jacketed.

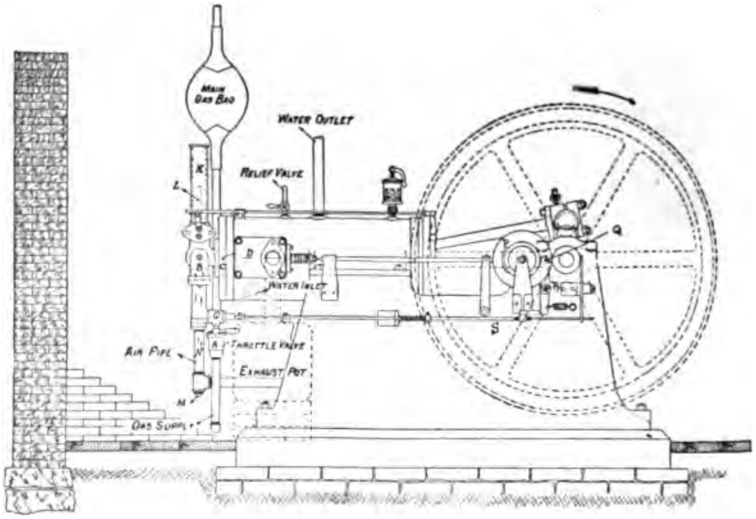

Fig. 11. Charter Engine. Side View.

The pressure in the cylinder when the exhaust valve opens is generally from 25 to 45 lb. above the atmospheric pressure, and the exhaust valve has to be lifted against this pressure. With a mushroom valve 4 in. in diameter, and with 40 lb. pressure per square inch at the end of expansion, there would be a total pressure of about 500 lb. on the valve at the time when it is to be lifted. It is desirable to reduce the strain on the valve and valve mechanism, and in large engines this is sometimes done by balancing the valve. A balanced exhaust valve *e* of the piston valve type is shown in Fig. 14 in its valve chest or housing. The connection with the cylinder is at *d* and the valve seat is the conical seat *a*. A hole *f* through the valve ensures the existence of atmospheric pressure on top of the valve, and the exhaust gases escape through *g* to the atmosphere. To prevent excessive heating of this valve water is circulated through it, entering at *b* and leaving at *c*.

In smaller engines the pressure on the exhaust valve just prior to its opening is sometimes relieved by the escape of the gases through an auxiliary exhaust port (P, Fig. 15) in the cylinder which is uncovered by the piston just before it reaches the end of its outstroke. Soon after it starts on the return stroke the piston covers the auxiliary port, and the exhaust for the remainder of the stroke is through the regular exhaust valve. As the regular valve (not shown in the figure) is not opened till after the uncovering of the auxiliary port, there is practically only atmospheric pressure on it when it lifts. An objection to this device is that the same auxiliary port is uncovered again near the end of the admission stroke, and as the pressure on the cylinder is then less than atmospheric pressure, some of the exhaust gases enter the

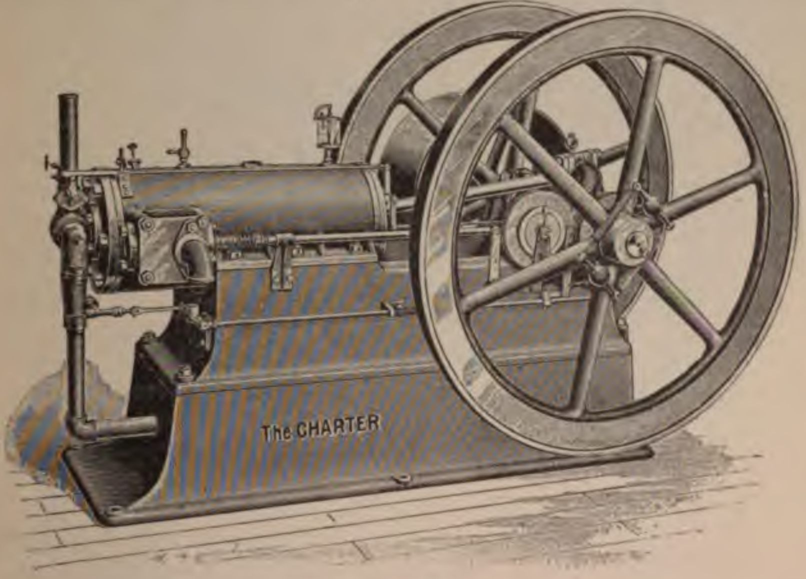

Fig. 12. Charter Engine.

cylinder, mixing with the charge and diluting it. At the beginning of the return or compression stroke part of the contents of the cylinder is forced out to the exhaust until the piston has again covered the auxiliary port, and consequently some of the charge is lost.

Valve Gearing. The valves are most commonly operated by cams. Cams are preferable to eccentrics for this purpose, because they can be designed to give very prompt opening and closing. The cams are mounted upon a *lay shaft*, or *side shaft*, or *cam shaft*. The cam shaft is driven in different engines either by spur gears, bevel gears, or skew gears. The spur gear (see Fig. 12) can be used only for parallel shafts, the bevel gear for shafts which are in the same plane but are inclined to one another, and the skew or spiral gear (Fig. 16) for shafts which are not parallel and do not lie in the same plane. To reduce the speed of the cam shaft the spur and bevel gears must have the gear on the cam shaft twice the size of that on the main shaft. With the skew gear there is no necessary relation between the diameter of the two gears, and generally the gear on the cam shaft is smaller than that on the main shaft. The skew gear has great advantage over the other two in its quietness of operation.

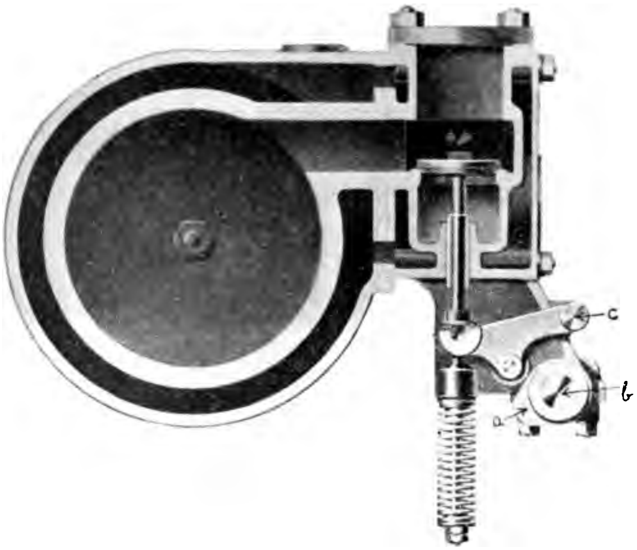

Fig. 13. Buffalo Engine. Cross-Section of Cylinder and Valve Chest.

THERMODYNAMICS OF THE OTTO CYCLE.

In internal combustion motors the explosive mixture in the cylinder consists of air mixed with a comparatively small volume of the gaseous or liquid fuel. For instance, if the engine uses gas from the city mains, the mixture will average about eight or nine parts of air to one of gas and should never have less than about six parts of air to one of gas. This mixture can be regarded up till the time when explosion takes place as if it were pure air. Also, the products of combustion, after the explosion is completed, have physical properties but very slightly different from those of air, and consequently the working substance in the cylinder can be regarded without serious error as consisting entirely of air. In the following discussion of what occurs in the engine cylinder, it is assumed throughout that the substance in the cylinder is air.

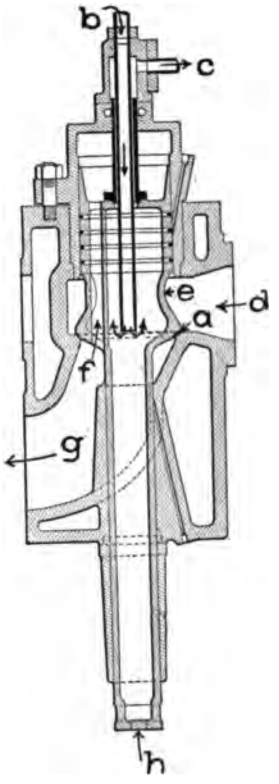

Fig. 14. Balanced Exhaust Valve.

The processes taking place in the engine cylinder are best represented on a pressure-volume diagram. At the beginning of the cycle of operations the piston is at the end of its path and is about to begin its out-stroke. The clearance space is full of products of combustion at atmospheric pressure because it has been in communication with the atmosphere through the exhaust valve which has just closed. The condition existing in the cylinder at this instant is represented in the diagram, Fig. 17, by the point 1, which is at a horizontal distance from the vertical axis, representing the clearance volume, and at a vertical distance above the horizontal axis, representing the atmospheric pressure of 14.7 lb. per square inch. As the piston makes its out-stroke, the admission valve opens, admitting the charge to the cylinder throughout the stroke, and as the cylinder is in communication with the outside air through the air admission valve, the pressure in the cylinder remains atmospheric pressure throughout the stroke. On the diagram the admission is represented by the line 1 2, which is at the constant height representing the atmospheric pressure and whose length represents the volume of the charge taken in, which is the same as the volume through which the piston moves. The point 2 represents the condition at the end of the first stroke. The admission valve now closes and the piston makes its return stroke. Since all the valves are closed, the charge cannot escape and is crowded into a smaller volume while its pressure rises. The process continues till the piston reaches the end of its stroke, at which time the whole charge is compressed into the clearance space. This process is represented by the line 2 3, which shows the rise in pressure resulting from the compression. A compression of this kind, occurring without the addition or the abstraction of heat from the gas, is called an *adiabatic* compression. It causes not only an increase in the pressure but also in the temperature of the gas. It is the process that takes place in the working of an ordinary bicycle pump and which causes its rise in temperature. The relation between the pressure of air and its volume when subjected to adiabatic compression is

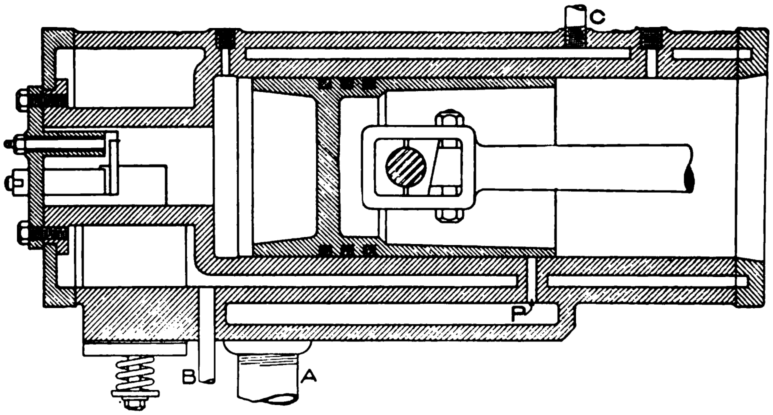

Fig. 15. Section Through Cylinder, Columbus Engine

$$PV^{1.405} = \text{constant}$$

(Note carefully that in this equation P means the absolute pressure and not the pressure shown by a gauge). When the charge has reached the conditions represented by the point 3, it is ignited, and the heat generated by the explosion raises the temperature and consequently the pressure of the mixture. The combustion occurs so rapidly that the piston has not time to start on the out-stroke before the combustion is completed and the rise of pressure occurs, as is shown by the line 3 4, while the volume of the gas is constant. The hot products of combustion at the pressure P_4 now force the piston out and, expanding behind it, they fall in pressure. This expansion, occurring without communication of heat to or from the gas is *adiabatic expansion*, and is consequently accompanied by a fall in temperature of the gas. The expansion curve 4 5 is similar to the compression curve 2 3, and has the same equation.

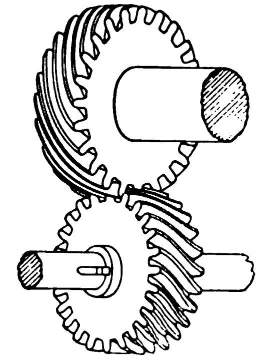
Fig. 16. Spiral Gear.

At the point 5 the piston is at the end of the stroke and no more expansion is possible. The exhaust valve opens and the pressure in the cylinder falls immediately to atmospheric pressure, as shown by the line 5 2 in the diagram. Throughout the last return stroke, 2 1, the exhaust valve remains open so that the pressure in the cylinder remains atmospheric pressure. The completed diagram, Fig. 17, shows the whole series of pressure and volume changes occurring in a gas engine, and is such a diagram as would be taken by an indicator from a perfect engine. The area 2 3 4 5 enclosed by the diagram represents the work done by the engine per cycle.

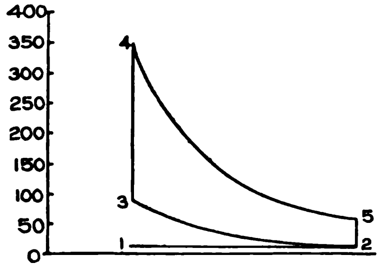

Fig. 17. Ideal Indicator Card of Otto Cycle.

The Pressures and Temperatures of the working substance and the amount of work done in an engine which exactly follows the Otto cycle can be readily calculated. Starting at the point

2 (Fig. 17), there is present in the cylinder a volume V_2 at atmospheric pressure P_2 and at the temperature t_2, which will be assumed to be the temperature of the air as it came into the cylinder. The working substance is compressed adiabatically till it fills only the clearance volume V_3. The consequent rise in pressure can be calculated from the formula already given, but it is more simply obtained from the curve, Fig. 18, which gives the relation between the changes of volume and of pressure in adiabatic expan-

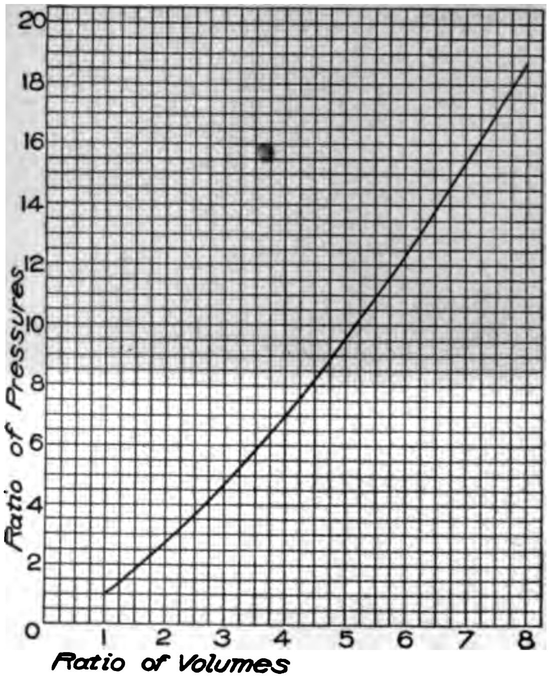

Fig. 18. Adiabatic Expansion.

sion or compression. The horizontal scale in this diagram is the ratio of expansion or compression, and the vertical scale shows the corresponding ratio of the pressures at the beginning and end of the expansion or compression. If, for example, the working substance expands adiabatically to five times its original volume, the pressure (which varies inversely as the volume) is shown by the curve to fall to $\frac{1}{9.67}$ of its original value. Conversely, if the working substance is compressed to $\frac{1}{5}$ the original volume, the pressure rises to 9.67 times its original value. Consequently, the pressure at the point 3, Fig. 17, can be found by the use of this curve.

Example. A gas engine with $33\frac{1}{3}$ per cent clearance takes in its charge at 14.7 lb. per sq. in. pressure. What is the pressure at the end of the adiabatic compression?

Solution. The clearance volume V_3 is $33\frac{1}{3}$ per cent of the volume, $V_2 - V_3$, through which the piston moves, or

$$V_3 = \frac{33\frac{1}{3}}{100}(V_2 - V_3)$$

$$\therefore \quad 3V_3 = V_2 - V_3$$

$$\text{and} \quad \frac{V_2}{V_3} = 4$$

From the curve, Fig. 18, if the ratio of compression is 4, the corresponding ratio of pressures is 7.06, so that the pressure at the end of compression is 7.06 times the pressure at the beginning of compression. Therefore the pressure at end of compression, $P_3 = 7.06 \times 14.7 = 103.8$ lb. per sq. in. abs.

The temperature at the end of the adiabatic expansion can be found from the equation for a perfect gas. This may be stated in the form

$$PV = wRT$$

where w is the weight of the gas, R is a constant for any perfect gas and has the value 53.2 for air, P is the pressure in lb. per sq. ft. abs., and T is the absolute temperature of the gas. The weight of the gas is constant throughout the adiabatic compression, and can be found from the point 2 if P_2, V_2 and T_2 are known. The temperature at 3 can then be found from the equation

$$P_3V_3 = wRT_3$$

Example. Assuming the conditions of the previous problem and supposing the temperature of the air to be 60° F., what is the temperature of the charge at the end of the compression?

Solution.

$$P_2V_2 = wRT_2$$

$$\therefore \quad wR = \frac{P_2V_2}{T_2} = \frac{14.7 \times 144 \times V_2}{60 + 461}$$

$$\text{Also} \quad T_3 = \frac{P_3V_3}{wR}$$

$$= P_3 \frac{V_3}{V_2} \times \frac{60 + 461}{14.7 \times 144}$$

$$\text{and} \quad \frac{V_3}{V_2} = \frac{1}{4}$$

$$\therefore \quad T_3 = 103.8 \times 144 \times \frac{1}{4} \times \frac{521}{14.7 \times 144}$$

$$= 919.6^\circ \text{ abs.}$$

$$\text{and } t_3 = 458.6^\circ \text{ F.}$$

The rise in temperature during explosion depends on how much heat is generated, which in turn depends on the strength of the explosive mixture and the heat of combustion of a cubic foot of the fuel. Let H be the heat of combustion of a cubic foot of the fuel in B. T. U., and let the mixture consist of 1 part of gas to n parts of air. The total volume of the charge taken into the cylinder each admission is

$$V_2 - V_1 \text{ cu. ft.}$$

the volume of fuel in this charge

$$\frac{1}{n + 1}(V_2 - V_1)$$

and the heat of combustion of this fuel is

$$Q = H \frac{V_2 - V_1}{n + 1} \text{ B.T.U.}$$

This heat is utilized in raising the temperature of the gas from the known temperature T_3 to another temperature T_4. The rise in temperature can be found when the heat necessary to raise one pound of air one degree in temperature is known. This amount of heat is called the *specific heat*. It is represented by the symbol C_v (indicating that the volume is unchanged while the temperature rises), and is equal to .169 B.T.U. for air. With a weight of w

lb. the heat necessary to raise the gas one degree in temperature is

$$wC_V \text{ B.T.U.}$$

To raise the temperature $T_4 - T_3$ degrees the heat supply is

$$w\,C_V\,(T_4 - T_3) \text{ B.T.U.}$$

and the heat of combustion is used entirely in raising the gas from T_3 to T_4.

$$\therefore \quad H\frac{V_2 - V_1}{n + 1} = w\,C_V\,(T_4 - T_3)$$

$$= \frac{P_2V_2}{RT_2}C_V\,(T_4 - T_3)$$

$$T_4 - T_3 = \frac{H}{n + 1} \times \frac{RT_2}{P_2} \times \frac{1}{C_V} \times \frac{V_2 - V_1}{V_2}$$

Example. In the previous problem if the charge takèn in consists of 1 part of gas to seven parts of air and the heat of combustion of the gas is 640 B.T.U., per cu. ft., find the temperature at the end of explosion.

Solution.

$$\frac{V_2 - V_1}{V_2} = \frac{V_2 - \frac{1}{4}V_2}{V_2} = \frac{3}{4}$$

$$T_4 - T_3 = \frac{640}{8} \times \frac{53.2 \times 521}{14.7 \times 144} \times \frac{1}{.169} \times \frac{3}{4}$$

$$\therefore \quad T_4 = 4649 + T_3$$

$$= 5568.6^\circ \text{ abs.}$$

$$\therefore \quad t_4 = 5107.6^\circ \text{ F.}$$

If a perfect gas is raised in temperature while its volume is unchanged, the absolute pressure will increase in exact proportion to the rise of absolute temperature,

or $$P_4 : P_3 :: T_4 : T_3$$

$$\therefore \quad P_4 = \frac{T_4}{T_3}P_3$$

Example. What is the pressure at the end of explosion in the preceding problem ?

Solution.

$$
\begin{aligned}
P_4 &= \frac{T_4}{T_3} P_3 \\
&= \frac{5568.6}{919.6} \times 103.8 \text{ lb. per sq. in. abs.} \\
&= 628.6 \text{ lb. per sq. in. abs.}
\end{aligned}
$$

The pressure and temperature at the end of the adiabatic expansion can be found most simply, after the other pressures and temperatures are known, by making use of a relation which exists between the pressures and temperatures at the points 2, 3 4. 5.* These relations are

$$\frac{P_2}{P_3} = \frac{P_5}{P_4}$$

and

$$\frac{T_2}{T_3} = \frac{T_5}{T_4}$$

Examples. What are (a) the pressures and (b) the temperatures at the end of the adiabatic expansion in the preceding problem ?

(a) $P_5 = \frac{P_2}{P_3} \times P_4 = 89$ lb. per sq. in. abs.

(b) $T_5 = \frac{T_2}{T_3} \times T_4 = 3155°$ abs.

$= 2694°$ F.

The work done by any heat engine is equal to the difference between the heat that goes to the engine and that which is rejected by the engine, because whatever heat disappears cannot have been destroyed and must have been converted into work. In the Otto cycle the heat taken in has been seen to be

$$Q = wC_V (T_4 - T_3) \text{ B.T.U.}$$

Heat is rejected from the engine only during the process represented by the line 5 2, because when the charge gets back to the condition 2, it has returned to its original volume and pressure and consequently to its original temperature. The heat rejected is then

---

* The ratio of the pressures $\frac{P_4}{P_5}$ can be obtained from the curve, Fig. 18, since the ratio of the volumes $\frac{V_5}{V_4}$ is known. But $V_5 = V_3$, therefore $\frac{V_5}{V_4} = \frac{V_2}{V_3}$ and $\frac{P_3}{P_2} = \frac{P_4}{P_5}$

$$Q_R = wC_v\,(T_5 - T_2)\ \text{B.T.U.}$$

And consequently the work done per cycle is

$$\begin{aligned} W &= Q - Q_R\ \text{B.T.U.} \\ &= 779\,(Q - Q_R)\ \text{ft. lb.} \end{aligned}$$

The efficiency of the cycle, that is, the fraction of the heat supplied that is converted into work is

$$\begin{aligned} E &= \frac{W}{Q} = \frac{Q - Q_R}{Q} \\ &= 1 - \frac{Q_R}{Q} \\ &= 1 - \frac{T_5 - T_2}{T_4 - T_3} \end{aligned}$$

And since, as already stated,

$$\frac{T_5}{T_4} = \frac{T_2}{T_3}$$

we get $$\frac{T_5 - T_2}{T_4 - T_3} = \frac{T_2}{T_3}$$

therefore $$E = 1 - \frac{T_2}{T_3}$$

Example. Find the efficiency of the cycle in the preceding problem.

$$\begin{aligned} E &= 1 - \frac{T_2}{T_3} \\ &= 1 - \frac{521}{919.6} \\ &= 1 - .567 = .433 \end{aligned}$$

The work W done per cycle can be calculated from the efficiency, without knowing the heat rejected

$$E = \frac{W}{Q}$$

or $$\begin{aligned} W &= E \times Q\quad \text{B.T.U.} \\ &= 779\,E \times Q\quad \text{ft. lb.} \end{aligned}$$

Examples. If the cycle discussed in the previous examples takes place in a cylinder of 12 in. diameter and 18 in. stroke, what will be the work done per cycle? If the engine makes 250 revolutions per minute, what will be its indicated horse-power?

Solution.

$$W = 779\ E \times Q \quad \text{ft. lb.}$$

$$Q = \frac{H}{n + 1}\,(V_2 - V_1) \quad \text{B.T.U.}$$

$V_2 - V_1$ is the volume through which the piston moves in cu. ft., and is the product of the cross section area of the cylinder in sq. ft. by the stroke in ft.

$$\therefore \quad V_2 - V_1 = \frac{\pi}{4} \times \left(\frac{12}{12}\right) \times \frac{18}{12}$$

$$= 1.178 \text{ cu. ft.}$$

$$\therefore \quad Q = 94.25 \text{ B.T.U.}$$

$$\therefore \quad W = 40.81 \text{ B.T.U.}$$

$$= 31791 \text{ ft. lb.}$$

Since this engine requires two revolutions to complete a cycle, the number of cycles per minute is only half the number of revolutions per minute; therefore the work per minute and the horse-power $= W \times 125$ ft. lb.

$$= \frac{31791 \times 125}{33000}$$

$$= 120.4 \text{ I.H.P.}$$

EXAMPLE FOR PRACTICE.

(a) A gas engine using the Otto cycle has 25 per cent clearance and takes in its charge at 14.7 lb. per sq. in. at 60 F. What is the pressure at the end of the compression?

Ans. 141.1 lb. per sq. in. abs.

(b) What is the temperature at the end of compression?

Ans. 539° F.

(c) If the charge consists of 1 part of gas to 9 parts of air and the heat of combustion of the gas is 600 B.T.U. per cu. ft. what is the temperature at the end of explosion? Ans. 4258° F.

(d) What is the pressure at the end of explosion?

Ans. 665.9 lb. per sq. in. abs.

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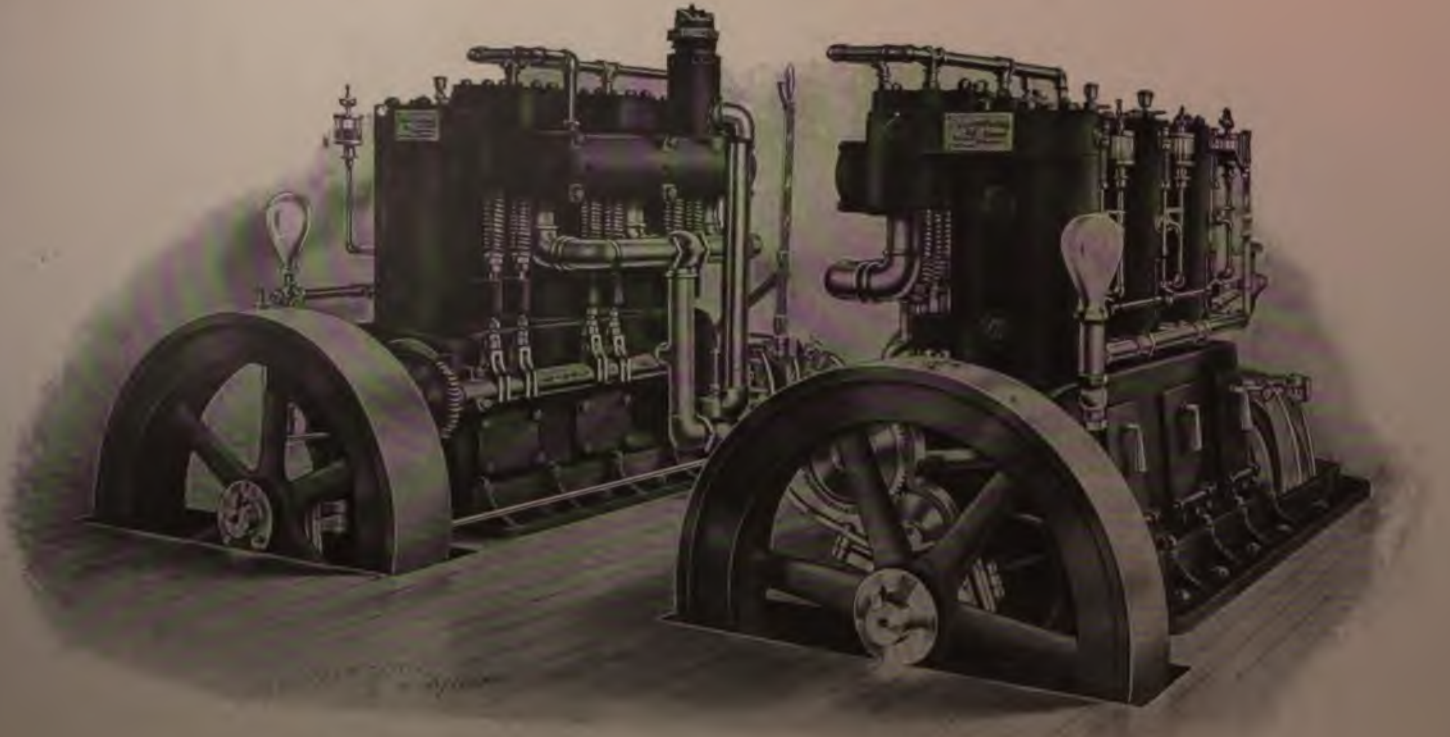


FORWARD VIEW OF A PAIR OF 27 H.P 3 CYLINDER 4 CYCLE BALANCED TWIN SCREW GAS ENGINES.
Wolverine Motor Works.

(e) What are the pressure and temperature at the end of the expansion? Ans. 69.4 lb. per sq. in. abs. 1997° F.

(f) What is the efficiency of the cycle? Ans. .479.

(g) If the cylinder diameter is 18 in. and the stroke is 24 in., and the engine makes 150 revolutions per minute, what is the I.H.P.? Ans. 180 I.H.P.

An examination of the equation for the efficiency of the Otto cycle,

$$E = 1 - \frac{T_2}{T_3}$$

brings out certain important results. The efficiency is seen to depend only on the ratio of the temperatures at the beginning and end of the compression, and not at all upon the temperature and pressure at the end of explosion. Since the ratio of the temperatures at the beginning and end of compression depends only upon the ratio of compression, and since further, the charge is always compressed till it occupies the clearance volume, the efficiency is seen to depend only upon the percentage clearance. In other words, in engines with the same percentage clearance using the Otto cycle, the percentage of the heat liberated in the cylinder that is converted into work is always the same whatever be the size of the engine or the strength of the charge. The effect of the clearance on the efficiency is exhibited in table I, where it is seen that the smaller the clearance the greater is the efficiency of the engine. The pressures at the end of compression are also given in the table, and are calculated on the assumption that the atmospheric pressure is 14.7 lb. per sq. in. abs.

TABLE I.

Percentage Clearance of Otto cycle engine.	Pressure at the end of compression lb. per sq. in. abs.	Efficiency of Otto cycle.	Efficiency of cycle with increased expansion, but with the same compression pressure as the Otto cycle.
20	183.3	51.6	60.9
25	141.1	47.9	58.4
30	115.4	44.8	55.
35	98.	42.1	52.5
40	85.5	39.8	50.4

) CYCLE WITH INCREASED EXPANSION.

ssure at the end of expansion is seen in the example
o be 89 lb. per sq. in. abs. In ordinary practice it is
und to be from 50 to 60 lb. abs. It is evident that
ere permitted to expand further it would do more
sequently would increase the efficiency of the cycle.
r card, Fig. 19, shows one method used for obtaining
nsion. The charge enters at atmospheric pressure from
n the admission is cut off. The piston continues moving forward to the end of its stroke, but as no more admission takes place the charge expands adiabatically to 3, while its pressure falls. On the return stroke the charge is compressed adiabatically, retracing the expansion path along 3 2 and continuing till the whole charge is compressed into the clearance space at 4. The rest of the cycle is unchanged. The diagram 1 2 4 5 8 2 represents the ordinary Otto cycle, and the shaded area 8 6 7 2 represents the increase in work due to the increased expansion. The efficiency of this cycle can be easily calculated and the results of such calculations are given in table I. They are made on the assumption that the charge is admitted for only one-half the stroke and that the heat combustion is 80 B.T.U. per cu. ft. of the charge. An inspection of the table shows the increase in efficiency which results from the increased expansion for engines which have the same pressures at the end of compression and indicates that in order that a gas engine of this type should be of high efficiency, it should compress the charge to a high pressure, and then should expand the products of combustion to a volume considerably in excess of the original volume of the charge.

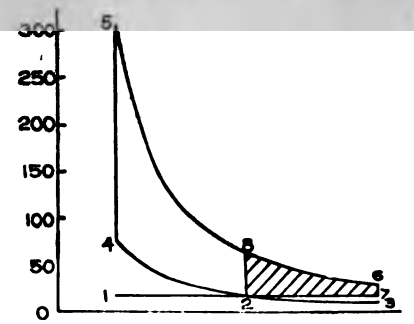


Fig. 19.

THE IDEAL AND THE REAL OTTO CYCLES.

The calculations in the preceding pages are made on the assumption that the gas engine follows the Otto cycle exactly, in which case the engine is called an *ideal* engine. The *real* engine does not exactly follow the Otto cycle because of certain practical difficulties. Differences between the real and the ideal engines occur in each part of the cycle. During admission (Fig. 20, line 1 2) the pressure in the cylinder is actually a pound or more below the atmospheric pressure, that difference being necessary to open the air admission valve (when automatic), and to cause the air to flow in with sufficient velocity. The charge, moreover, is heated by contact with the cylinder walls and with the hot gases remaining in the clearance. The compression is not adiabatic because it occurs in a cast iron cylinder which takes heat from the gas while it is being compressed and so makes the final temperature and pressure less than that calculated on the assumption of adiabatic expansion. The difference, generally, is not very great.

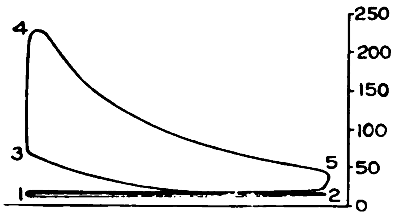


Fig. 20. Indicator Card from Otto Cycle Engine.

The explosion in the real engine is neither instantaneous nor complete. It approximates more closely to the ideal explosion when the compression is considerable and when the explosive mixture has only a small excess of air present. With weaker mixtures the explosion becomes slower and less complete, as shown in Fig. 21, till, with the weakest explosive mixture, the process is really one of slow combustion taking place throughout the whole of the expansion period, and some of the charge may be still unburned when exhaust takes place. Even under the best conditions the rise of temperature, and consequently of pressure, during the explosion is only about six-tenths of that occurring in the ideal engine. This, it will be seen, makes the power of the real engine considerably less than that of the ideal. The water jacket around the cylinder, without which the cylinder would be too hot to be properly lubricated, is one of the

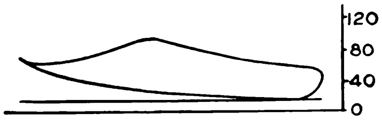


Fig. 21. Indicator Card with Weak Mixture.

auses of the difference between the real and ideal cycles, as t absorbs usually about forty per cent of the total heat of oustion.

xpansion curve is above the adiabatic in real engines cylinder walls that have been heated by the explosion some heat to the gases and also because the combustion s continues and liberates more heat. This last effect is espec marked when the explosive mixture is weak.

Finally, the exhaust, as in the steam engine, begins a little before the end of the expansion stroke so as to give plenty of time for the escape of the gases, and the pressure in the cylinder during the exhaust stroke is necessarily higher than that of the atmosphere into which the gases are rejected.

The total effect of all these differences between the real and the ideal engine is that the work done in an actual engine in good condition is only from five-tenths to six-tenths of that which the ideal engine would do, and the efficiency of the real engine is only from five-tenths to six-tenths of that of the ideal engine.

Example. What are the probable actual efficiency, horse-power and gas consumption of the engine whose ideal performance has been worked out in the preceding examples? Assume the real engine to have $\frac{6}{10}$ the efficiency of the ideal engine.

The ideal efficiency was found to be .433.

$\therefore$ The probable real efficiency $= .6 \times .433 = .26$.

The ideal horse-power was found to be 120.4,
the probable real H. P. $= .6 \times 120.4 = 72.2$.

The gas consumption is expressed in cu. ft. per I.H.P. per hour. In the ideal engine the volume of gas taken in per cycle was

$$\frac{V_2 - V_1}{n+1} = \frac{1.178}{8} = .147 \text{ cu. ft.}$$

The number of cycles per minute was 125.

$$\begin{aligned} \therefore \text{ the gas used per minute} &= .147 \times 125 \text{ cu. ft.} \\ &= 18.4 \text{ cu. ft.} \\ \therefore \text{ the gas used per hour} &= 18.4 \times 60 \text{ cu. ft.} \\ &= 1104. \text{ cu. ft.} \end{aligned}$$

And the probable real I.H.P. is 72.2.

$$\therefore \text{ the gas used per I.H.P. per hour} = \frac{1104.}{72.2} = 15.3 \text{ cu. ft.}$$

EXAMPLE FOR PRACTICE.

What are the probable actual efficiency, I.H.P. and gas consumption of the engine whose ideal performance has been worked out in the previous examples for practice.

Answers: .287 efficiency.
108 I.H.P.
14.71 cu. ft. gas consumption.

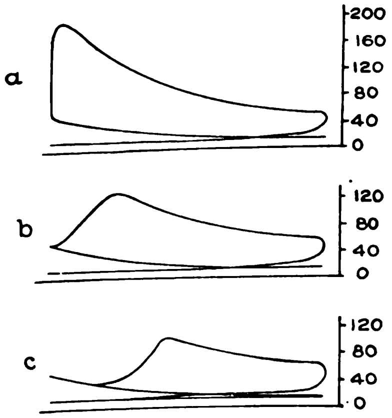


Fig. 22.

Ignition. For satisfactory action of a gas engine the ignition of the explosive mixture must be certain and must occur at a definite predetermined time. In *timing* the ignition it has to be recognized that the explosion is not instantaneous but requires the lapse of a not inconsiderable period of time before the maximum pressure is reached. The actual duration of the explosion depends on the strength of the explosive mixture and on the amount of compression to which it is subjected. The ignition should have *lead*, that is, should begin a little before the end of the return or compression stroke, when the crank is about 15 from its dead center, so that the maximum pressure is reached when the crank has just passed the dead center. The indicator card, *a*, Fig. 22, is with properly timed ignition. If the ignition is later than this, indicator cards similar to *b* or *c* will be obtained, and the engine will do less work and be less efficient. If the ignition is too early, the maximum pressure will be obtained (Fig. 23) before the crank has reached its dead center and will tend to reverse the engine. This causes great shock to the engine, its rapid deterioration and low-

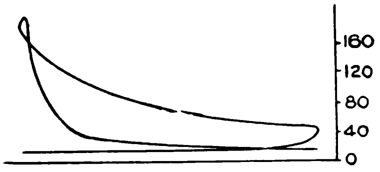


Fig. 23.

ered efficiency. The immediate external evidence of too early or premature ignition, from whatsoever cause, is a violent pounding noise in the engine.

Two methods of ignition are in common use in engines using the Otto cycle. The first is by bringing the explosive mixture into contact with some surface which is kept at a temperature sufficiently high to cause ignition; the second is by means of an electric arc. A hot tube is the common device when the first method of ignition is used. The tube E (Fig. 24) is closed at the upper end and communicates at its lower end through the port B with the cylinder A. It is heated by an external flame from the Bunsen burner C, and is maintained at a full red heat. The chimney around the tube is lined with asbestos and keeps the flame in good contact with the tube. During the admission stroke the tube is filled with products of combustion at atmospheric pressure remaining from the previous explosion. As compression goes on, the non-explosive products of combustion are crowded into the upper part of the tube while part of the explosive mixture in the cylinder is compressed into the lower part of the tube. The length of the tube and the position of the flame are adjusted by experiment so that the explosive charge will just reach the hot portion of the tube and be ignited at the moment when ignition is desired. Shortening the tube makes the ignition come earlier. With this device the actual time of ignition is not very definite. It depends on the temperature of the tube, the position of the Bunsen flame, the strength of the

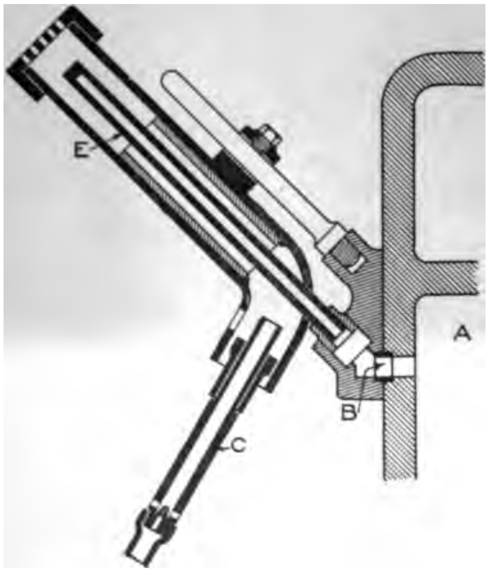


Fig. 24 Hot Tube Igniter.

mixture and the amount of compression. As these last two quantities are purposely varied by the governor in some engines, irregular timing would result from its use in such cases.

The irregularity of timing with the hot tube igniter can be partly remedied by the use of a *timing valve*. The timing valve B (Fig. 25) is held on its seat by a spiral spring D until ignition is desired, when by a movement of the bell crank lever E the valve opens and the compressed charge in the cylinder A gets access to the hot tube C. The valve B is kept open till the end of the exhaust stroke. The tubes are preferably made of nickel alloy or of porcelain, but the latter is very brittle and apt to break when being fastened in place. Iron tubes are used sometimes, but they burn out rapidly and are unreliable.

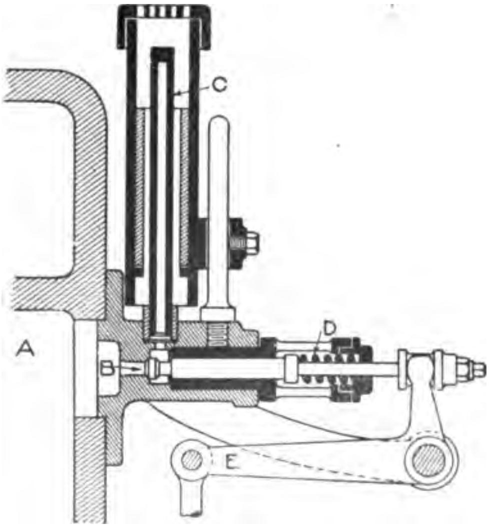


Fig. 25. Hot Tube Igniter with Timing Valve.

Even when provided with a timing valve the hot tube does not give very satisfactory ignition; and, moreover, some time is consumed in heating the tube before the engine can be started. Accurate timing can be obtained best by electric means, and *electric ignition* is consequently used more than any other. The method is to make a spark pass at the instant when ignition is desired between two terminals situated in the clearance space of the engine. The most common way of forming the spark is to separate two contact points through which a current has been flowing. An electric arc will then pass between the separating contact points. In order to ensure that the temperature of the arc is high enough and its duration is sufficient to ignite the explosive mixture through which it passes, a *spark coil* is generally inserted in the circuit. A spark coil consists merely of a bundle of soft iron wires surrounded by a coil of insulated copper wire, through which the current goes. The contact points of the igniter must be brought together to re-establish the current before another spark can be obtained. A device of this nature is known as a *make-and-break* igniter; and when the contact points do not slide across one another, it is called a *hammer break* contact.

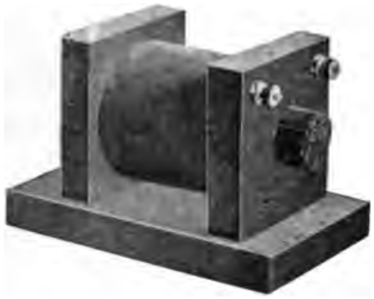


Fig. 26. Spark Coil.

One of the common forms of hammer break igniter is illustrated in Fig. 27, which shows an igniter plug removed from the cylinder head. The movable electrode *b* is at the end of an arm which is fastened to the spindle *e*. When the interrupter lever *d*, which is loose on the spindle *e*, and is connected to it through a coiled spring, is lifted by an arm from the cam shaft of the engine, it rotates the spindle *e* so as to bring *b* into hard contact with the stationary and thoroughly insulated electrode *a*. This completes a circuit and permits a current to flow from *a* to *b*. When ignition is desired the lever *d* is tripped and flies back, carrying with it the shaft *e*, abruptly breaking the contact and causing an electric arc to form between *a* and *b*. The contact points are generally made of platinum, as this does not oxidize or corrode, but other metals are also used. The passage of the spark takes minute particles of the material from one terminal and deposits them on the other, the action following the direction of the current. By reversing the direction of the current, the material may

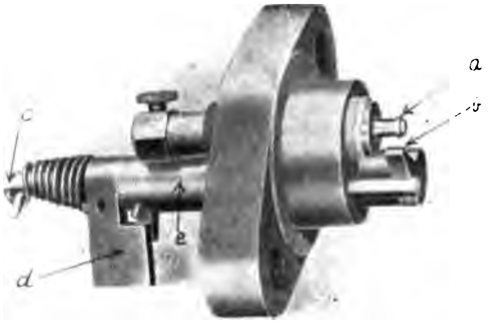


Fig. 27. Otto Engine Igniter Plug.

be returned to the terminal from which it was taken, and the durability of the contact points considerably increased.

The current is generally taken from a primary battery, consisting of about five cells. The Edison-Lalande cell, made up of two zinc plates and a plate of compressed copper oxide immersed in a strong solution of caustic soda, is perhaps the most largely used. Other sources of electricity can be used. Current is sometimes taken from a direct current lighting or power circuit, but this is objectionable because the circuit is grounded every time the igniter terminals are in contact. The practice is growing of using a small special dynamo for the exclusive purpose of supplying the current for ignition. This makes the ignition spark more certain and of more uniform strength than when a battery is used, as the latter deteriorates and weakens with use.

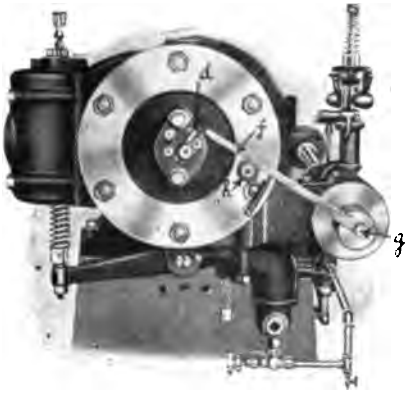


Fig. 28. Igniter Gear of Root and Van Dervoort Engine.

A make-and-break contact is sometimes obtained by sliding one contact point over the other until it slides off completely. This is known as a *wipe break*. The method ensures a good contact, produces a very hot spark, keeps the contact points clean, but wears them out quite rapidly. Provision must be made for adjustment, otherwise the timing will alter with the wear of the points. The rubbing surfaces can be of iron.

The igniter gear of an engine with hammer break ignition is shown in Fig. 28. The igniter rod *f*, which is supported on the reel *h*, receives a reciprocating motion from a crank *g* at the end of the side shaft. During the exhaust or admission stroke the end of the rod *f* comes in contact with the interrupter lever *d* (compare with Fig. 27) and establishes the contact of the electrodes. The vertical component of the movement of the end of the rod *f* sets free the lever *d* at the moment when ignition is desired.

A switch (Fig. 29) should always be included in the electric circuit and should be thrown out when the engine is not running, so as to prevent the short-circuiting and consequent exhaustion of the batteries.

Another way of obtaining electric ignition is known as the *jump spark* method. In this system the terminals are stationary, generally from one-sixteenth to one-eighth of an inch apart, and the spark is made to spring across the gap between them by putting the terminals in the secondary circuit of a Ruhmkorff or *induction coil.* This coil consists of a core of soft iron wire around which is wound a relatively coarse insulated wire, the *primary circuit*, through which the current from the source of energy flows. A relatively fine insulated wire coil, the *secondary circuit*, is wound around the primary coil, but has no metallic contact with it. If the current flowing through the primary circuit is varied in strength, it creates or induces a current in the secondary circuit. Generally the Ruhmkorff coil is provided with a magnetic vibrator (similar to that used in an electric bell), which makes and breaks the primary circuit with great rapidity and induces a considerable alternating current in the secondary circuit. If the two ends of the secondary coil be brought close to one another, but not quite in contact, a spark will jump across the gap at each make and break of the primary current, the spark at the break being the more powerful. For ignition of the explosive mixture in a gas engine it is not necessary to use a vibrator; the cam shaft of the engine breaks the primary circuit at the instant when explosion is desired. The spark passing on the subsequent

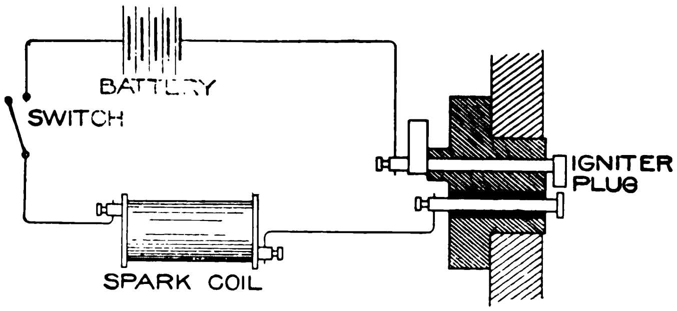


Fig. 29. Diagram of Igniter Circuit.

remaking of the primary circuit is not strong enough to ignite the charge. The connections for jump spark ignition are shown diagrammatically in Fig. 30. The primary circuit is shown there as being completed through a cam on the side shaft of the engine. As soon as the side shaft has moved from the position shown, the contact with the upper flat spring is broken and the primary current is interrupted, thereby inducing sufficient current in the secondary circuit to make a spark pass across the air gap between the terminals of the spark plug. These terminals are completely insulated, so that the only path for a current between them is across the air gap.

The jump spark method has as its great advantage the absence of moving parts inside the cylinder. This is offset by the fact that the spark is liable to fail as a result of the formation of deposits of rust or corrosion on the points, the liability to this being much greater than in either of the make-and-break methods. The difficulty of obtaining satisfactory insulation is also greater.

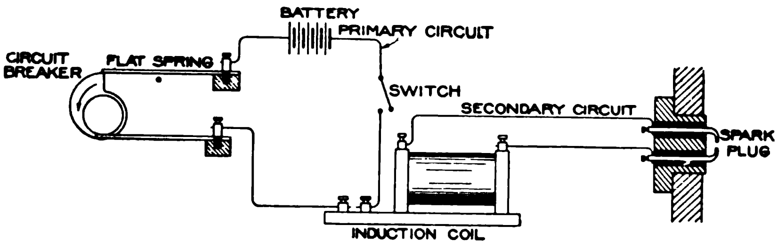


Fig. 30. Diagram of Jump Spark Igniter and Connections.

Governing. The governing of an engine means the control of the power which it is developing so that its speed is maintained practically constant. If the engine develops more power than is required, the engine will speed up; if the power delivered to the crank shaft is less than the resistance there, the engine will slow down. The governing of a gas engine, like that of the steam engine, is effected by utilizing small variations of engine-speed resulting from change of engine load. The controlling mechanism, or the governor proper, does not differ from that used on the steam engine, but there is a considerable difference in the way in which it controls the work done by the engine. There are two general methods in use in gas engines for varying the power; one by varying the number of explosions or impulses per minute, which is known as the *hit-and-miss* system, and the other by varying the magnitude of the impulse while keeping the number per minute constant, which may be called the *variable impulse* system.

The Hit-and-Miss System. The omission of the explosion or impulse can be obtained in several ways. The most common method is to keep the gas admission valve closed so that air alone is taken in during the admission stroke, and consequently there is no explosion. A method of accomplishing this is to be seen in Fig. 31, in which a loaded centrifugal governor is shown driven by bevel gearing from the cam shaft. In the position shown, the gas admission cam *d* will come under the reel *c*, and will start to lift it at the beginning of the admission stroke. The reel *c* is loose on a spindle at the end of the horizontal lever *e*, and the vertical rise of the spindle due to the action of the cam opens the gas valve by a system of levers not shown in the figure. If the engine speeds up, the rise of the governor balls raises the sleeve on the governor spindle, lifts the horizontal arm of the bell crank lever fulcrumed at *a* and shifts the forked end *b* of the vertical arm to the right, carrying the reel *c* with it, so that the cam no longer engages it and no gas is admitted. When the speed comes down to the normal speed, the reel is moved back and the admission of gas again takes place. This method is open to the objection common to all the hit-and-miss methods that it makes the speed of the engine very irregular at any other than full load. Even at full load, with the Otto cycle occurring in a single acting cylinder, there is only one motive stroke or impulse in four strokes instead of one every stroke as in a double acting steam engine. If the engine governs by the hit-and-miss method and is running at half load, half the explosions will be omitted and there will be but one motive stroke in eight; at one-third load, there is but one motive stroke in twelve; and at

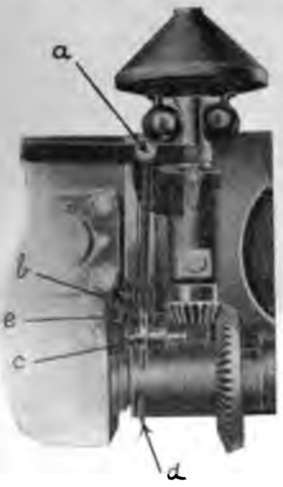


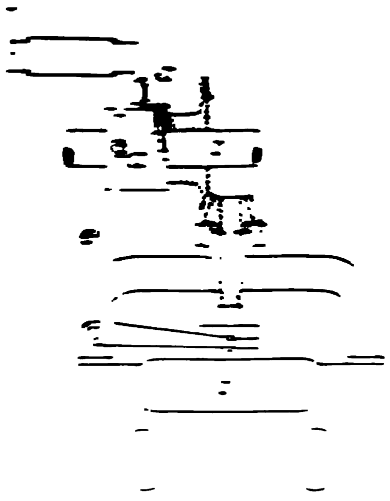
Fig. 31. Governor of Otto Engine.

quarter load, one in sixteen. Running at quarter load, the engine will be speeded up during the motive stroke and will slow down during the succeeding fifteen strokes till it gets to normal speed again. The actual variation in speed at low loads can be reduced by the use of heavy fly wheels, but with this method of governing it is too great for use when close regulation is necessary, as for electric lighting. There is an incidental advantage in the use of this method in that, during the idle cycles, the cylinder is flushed out by the *scavenging* charge of air, which makes the next explosion more powerful.

The omission of an explosion is sometimes effected in engines which have an automatic admission valve by the action of the governor in keeping the exhaust valve open throughout the cycle. The free communication between the cylinder and the outside through the exhaust valve prevents the pressure in the cylinder from falling sufficiently below the atmospheric pressure, during the admission stroke, to cause the inlet valve to open. The cylinder contains only products of combustion, substantially at atmospheric pressure, so long as the exhaust is open, and consequently no explosion can occur.

The Variable Impulse System. The amount of work done in a given gas engine depends on the strength of the charge, on its amount, on the timing of the ignition, and on several other factors. The engine can be governed by the variation of *any one* of these, and the three specifically mentioned are all in regular use for this purpose.

If the governing is effected by *varying the strength of the charge*, the control has to be such that the mixture is always an explosive one. With each kind of gas used in an engine there are both higher and lower limits to the amount of air with which it may be mixed if it is to remain an explosive mixture. If the ratio of air to gas should be outside these limits, the mixture sent to the exhaust would be unburned and valuable gas would be lost. Conseqently, if the engine goes above normal speed when admitting the weakest explosive mixture, the power of the engine has to be further reduced by omitting the admission of gas entirely. In Fig. 32 is shown a device for governing in the manner just described. The governor *d* is driven from the cam shaft *c* through the [illegible] [illegible] [illegible] in section. This is accomplished by raising the [illegible] of the [illegible] on which is a [illegible] similar to [illegible] in Fig. 31. The [illegible] is free to slide on a feather on the cam shaft, its exact position being controlled by the governor through the bell-crank [illegible]. On the sleeve [illegible] is a series of cams of the same throw [illegible] different [illegible] lengths. The function of the [illegible] of the [illegible] by shifting the sleeve so as to bring different cams into engagement with [illegible]. In the position shown the engine is [illegible] normal speed, and the sleeve is at extreme position to the right and no gas is being admitted. As the speed of the engine falls, the sleeve moves to the left, admitting gas for a definite period for each engine speed. With full load on the engine, the feed engages with the longest cam and the strongest mixture is admitted to the cylinder.



This [illegible] method of govern- [illegible] [illegible] of the [illegible] were taken into [illegible] [illegible] [illegible] [illegible] the same. The [illegible] [illegible] larger than for [illegible] of Figs. 30 and 31 [illegible] power of the engine. [illegible] strength of the [illegible] running of [illegible] be able to run the engine with a [illegible] when a hit-and-miss [illegible] When it is desired to have an impulse every [illegible] a constant strength of mixture can be maintained if the *power of the engine is controlled by varying the amount of the mixture taken in.* An example is shown in Figs. 33 and 34 of the actual mechanism used for this purpose. Gas from the passage

G enters a port in the cylindrical valve A and meets air which enters from D through similar ports. The mixture passes out of the valve through a large port near the top and goes through C to the cylinder where the inlet valve D is open. The relative amounts of gas and air are regulated by the two levers H H, which are connected to entirely independent cylindrical shells inside the valve A and which can rotate them so as to cover up more or less of the lengths and therefore of the areas of the gas and air ports respectively. With the two levers in constant positions the areas for admission of gas and air to the cylindrical valve A will be fixed, and consequently the strength of the mixture will be constant. The actual amount of the mixture entering the cylinder is controlled by the governor B, which works an internal cylindrical valve in such way as to throttle the discharge port of the valve A when the speed increases.

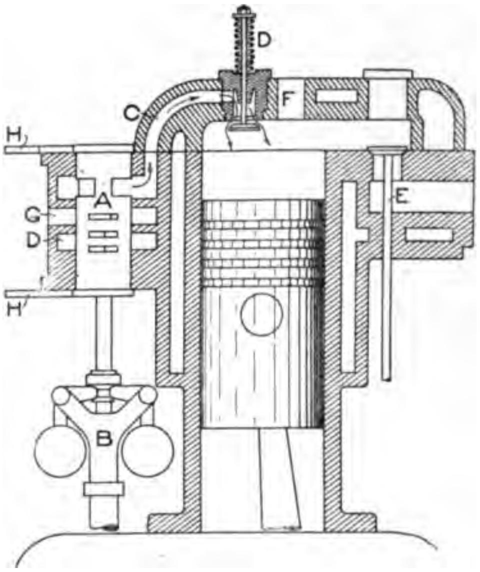


Fig. 33. Mixing Valve of Westinghouse Engine.

This method of governing permits a perfect adjustment of the work done in the cylinder each cycle, and consequently gives more uniform speed of the engine than any of the methods so far described. The throttling of the mixture imposes extra work upon the engine during the admission stroke, as the piston has to move out with a vacuum behind it. At the end of the admission the pressurė in the cylinder will be less and less as the load on the engine becomes smaller, and consequently the pressure in the cylinder at the end of compression is less as the load decreases. With decreased compression the combustion of the mixture is slower. This is well shown in Fig. 35, which gives a series of indicator cards taken at different loads from an engine using a strong mixture and a throttling governor.

Another method of accomplishing the same result is to admit a mixture at atmospheric pressure for part of the admission stroke only, the duration of the admission being determined by the governor. This method of governing gives an indicator card similar to Fig. 19. The difference between an engine governing in this way and one governing by the throttling method is similar to that between a Corliss steam engine and a throttling steam engine. The advantage of *cut-off* governing is in the decreased work done by the engine in drawing the charge into the cylinder. The use of a partial charge, whether obtained by throttling or by cutting-off, permits the expansion of the exploded mixture to a lower pressure than is possible in an engine using a full charge and having the same percentage of compression. This is the practical method of obtaining the increased expansion, the advantage of which has been already pointed out.

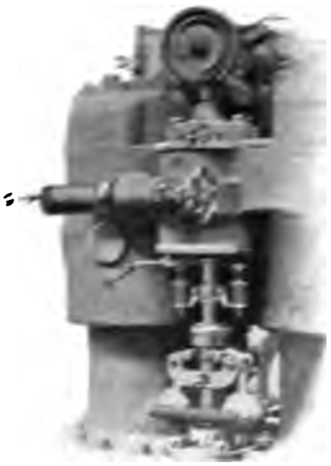


Fig. 34.—Governor of Westinghouse Engine.

When economy is not of the greatest importance, as, for instance, in automobile practice, the power of the engine may be controlled by *varying the point of ignition*. It has been shown already (Fig. 22) that the power of the engine decreases as the lead of the ignition becomes less. If the ignition occurs after the beginning of the stroke, the lead is said to be negative, and the power is greatly decreased. If the lead is increased (Fig. 23), there still results a decrease of power. The

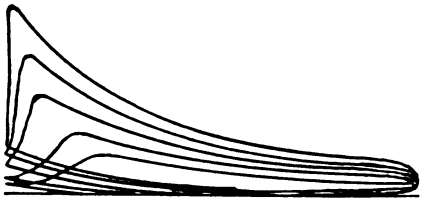
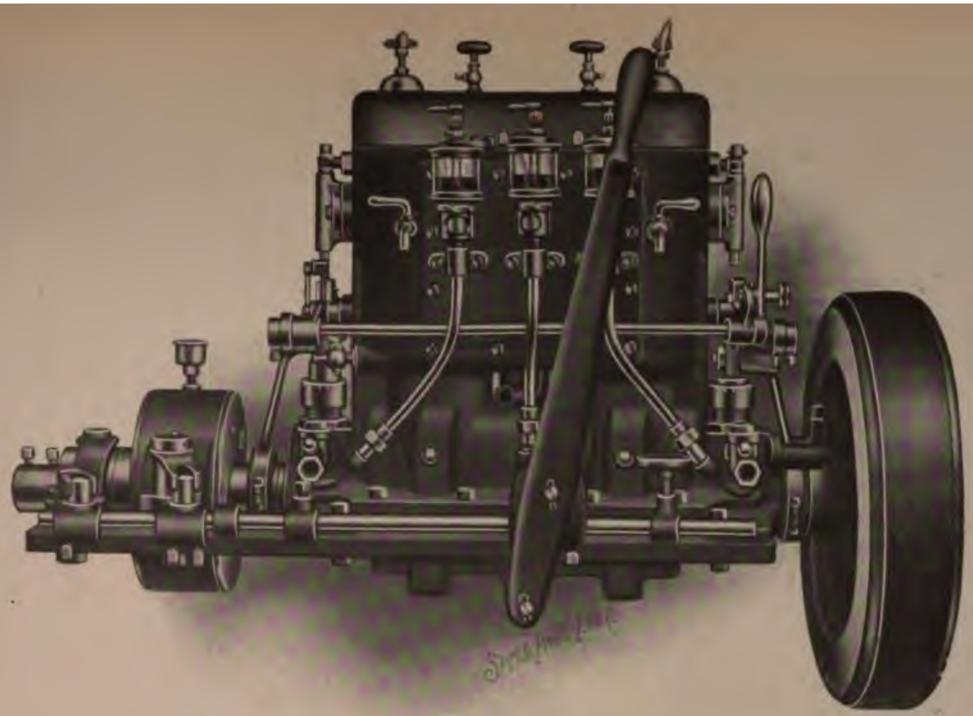


Fig. 35.



TOQUET MARINE MOTOR.
Starboard Side with Reverse Gear.

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TILDEN FOUNDATIONS

control of the power by varying the ignition is always uneconomical, but the method is one of extreme simplicity.

Starting. A gas engine will not start itself in the way a steam engine does when steam is turned on. It is necessary to get the engine in motion by means of some special source of power before it can take up its normal cycle of operations. Generally this special source of power is not adequate to get the engine moving rapidly when it is connected to any considerable load; it is always preferable and generally necessary to throw the load completely off the engine till it gets under way.

In the normal running of an engine the ignition of the charge occurs before the end of the back stroke, and if the time of ignition is kept the same when starting there is a danger, or often the certainty, that the high pressure of the explosion, acting on the piston before the crank has got to its dead center, will overcome the inertia of the engine, which is small because of its low speed, and will reverse its direction of rotation. If the starting power is small, the ignition has to be retarded by some special device so that it will not occur till after the crank has passed its dead center —that is, there must be negative lead. An example of a device for retarding the ignition is given in Fig. 36. The igniter rod A (compare with *f*, Fig. 28) which is worked by a crank on the side shaft B, is supported during normal running on the reel C, which is loose on the fixed spindle D. In the position shown it is just about to trip the interrupter lever E on the spindle which moves the movable electrode. When starting, the reel C is slid along the spindle D so that the igniter rod A rests, as shown in the dotted lines, directly on D, consequently the tripping occurs so much later.

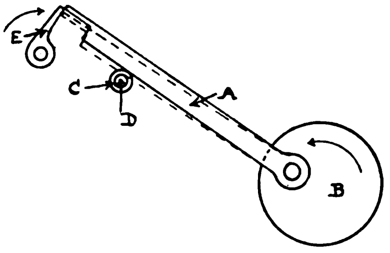

Fig. 36.

There are several general methods of starting gas engines. If the engine is small, not exceeding 10 H. P., and can be disconnected from its load, it is common to start it by turning it over by hand for a few revolutions till an explosive mixture is admitted and ignited. As it is difficult to pull the engine over when the charge is compressed for the whole back stroke, most engines are provided with an extra exhaust cam which is put into action while starting, and which not only opens the exhaust valve during the exhaust period but also opens it again during the first part of the compression period, so that some of the explosive mixture is forced out of the cylinder and the amount of compression is decreased. The explosion of this diminished charge after the crank has passed the dead point, starts the engine going, and after operation under these conditions for several cycles, the engine will come up to speed if it is not loaded heavily, and the compression and ignition may then be changed back to the normal running conditions.

With large engines it is impracticable to start by hand, and other devices have to be used. One of the most simple and certain is to start the engine by the admission of compressed air, which acts on the piston just as steam does in a steam engine. This method is especially desirable in an engine with several cylinders, in which case one cylinder is used as a compressed air cylinder to run the engine till the other cylinders take up their normal cycle of operations; and then the compressed air is shut off and the first cylinder is put into normal action. If the engine has only one cylinder, it can be brought to a good speed by the admission of compressed air, and then, after the compressed air is shut off, it will continue to revolve by its own inertia until an explosive mixture is taken in and exploded.

An arrangement for starting a multicylinder engine with compressed air is illustrated in Figs. 3 and 37. A compressor (Fig. 37), which is driven by a belt from the engine, forces air into a storage tank and brings it to a pressure of about 160 lb. In case of need the compressor can be operated by hand. When the engine is to be started, the compressed air can be admitted to one of the cylinders. The cam B (Fig. 3) on the upper shaft is first thrown out of action by a special device, so that the inlet valve J cannot open. The hand lever on the outside of the crank case near the cam A is thrown over, putting the ordinary exhaust cam A out of action, but bringing into action a double cam which keeps the exhaust valve E open throughout every up-stroke of the engine. Another cam on the same shaft is brought into action at the same time, and operates a starting valve (not shown) on the pipe from the compressed air reservoir, admitting compressed air to the cylinder on every down stroke. The cylinder then acts as a compressed air engine till the explosions begin in the other cylinders, when the cams B and A are brought back to their normal positions and the starting cylinder functions normally. In other engines compressed air is admitted to the cylinder during the expansion stroke by manual operation of a special valve. After two or three admissions during successive cycles, the engine will attain speed enough to permit the opening of the gas valve and the commencement of the cycle.

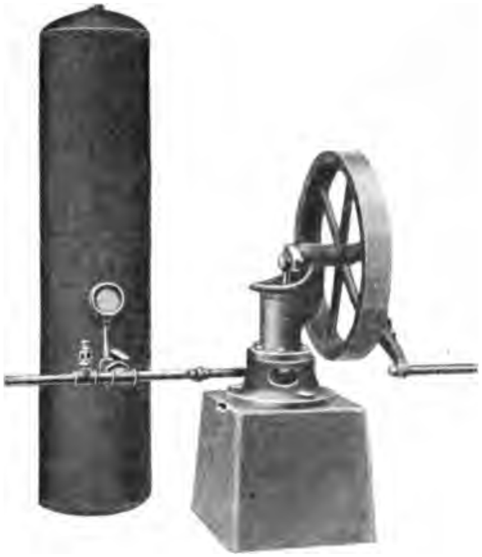

Fig. 37. Air Compressor.

With engines up to 100 H.P. a common method of starting is to ignite a charge which has been drawn into the engine by turning it over by hand. The engine is brought to the beginning of the expansion stroke, and a definite amount of gasoline is put into a cup which connects with the cylinder through a valve which is opened. The engine is then pulled over till the piston has made half its forward stroke, air being drawn in and forming an explosive mixture with the gasoline which enters at the same time. The gasoline valve is then closed and the engine is turned quickly in the opposite direction, the charge is compressed as much as possible, and is then ignited. The ignition is brought about by tripping the electric igniter by hand, or by the use of a special detonator, or even, in some cases, by striking a match inside the cylinder by means of a special device. It is not possible with a loaded engine to compress the charge much by hand, so that this method is only applicable to engines of moderate size which can be disconnected from their starting load.

If the engine has to start under moderate load, it is generally necessary to supply the engine with a charge which has been compressed to a high pressure. This can be accomplished by setting the engine with the crank about ten degrees past the dead center on the expansion stroke, and then pumping an explosive mixture into the cylinder (Fig. 38) till the piston begins to move. At that instant the charge is ignited, and the work done by the expansion of the exploded charge will be enough to start the engine on its cycle of operations. Another method of accomplishing the same thing is to connect the cylinder E (Fig. 39) with a special starting chamber D. When the engine is being shut down, the special inlet valve A is lifted from its seat, so that at each suction stroke air is drawn through the chamber D by way of the valve F. The chamber D, the cylinder, and the connecting pipe are thus filled with pure air at atmospheric pressure. When the engine is to be started the gas cock C is opened and gas flows both into the chamber D and into the cylinder, a cock on the cylinder being opened. A pilot light burns across the opening above the valve F, and after a

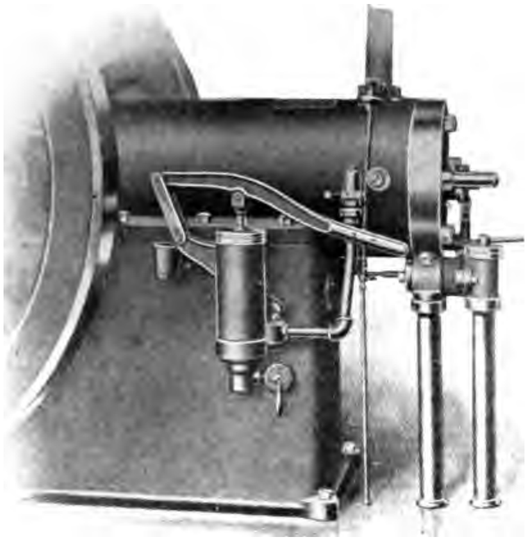

Fig. 38. Starting Gear of the Fairbanks-Morse Engine.

short time a combustible mixture of air and gas issues and catches fire. If the cock C is then closed, the flow of the explosive mixture stops and the flame consequently shoots back past the valve F and ignites the mixture in D, closing the valve F against an upper face by the force of the explosion. The flame proceeds to the cylinder, the contents of which will have been compressed by the explosion in D, and causes an explosion there. In large plants a special starting engine is customary.

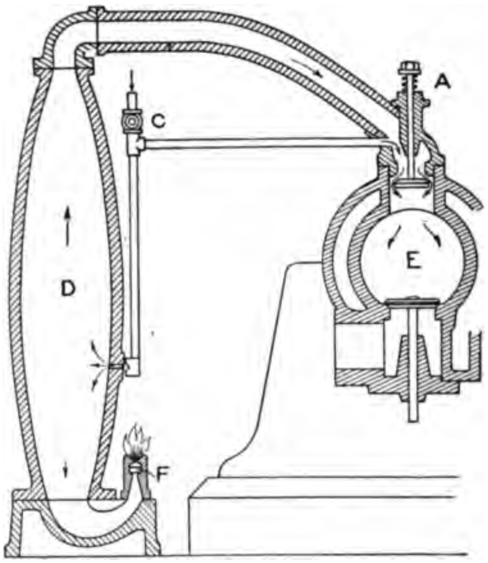

Fig. 39.

Water Jacket. In all the preceding sectional views of gas engine cylinders it will be seen that the cylinder barrel and the cylinder head have double walls and in every case provision is made for the active circulation of water through the space between the two walls. Without the use of a *water jacket*, or some equivalent device, the engine would be inoperative, because the high temperature to which the cylinder would be raised by the explosions would not only vaporize the lubricating oil and cause the rapid destruction of the cylinder, but also the entering mixture would be exploded before its time by contact with the hot metal. The necessity for effective cooling is greater in the larger engines; it is often necessary to water-jacket the exhaust valve in large engines that it may not be warped out of shape by the high temperature, and may not be hot enough to ignite the entering charge. The cooling arrangement for a balanced exhaust valve is shown in Fig. 14, the water entering the valve through the tube *b* and escaping after circulating at *c*. In some large engines the pistons also are water-jacketed. In very small engines it is po[illegible], when the engine is placed in a strong current of air, to replace [illegible]e water jacket by a system of thin metal ribs (Fig. 40), or points, [illegible] the external surface of the cylinder. The current of air can be obtained either from a fan driven by the engine, or, as in bicycle motors, by the movement of the engine itself. When the engine is water-jacketed it is often practicable with small engines to use the same cooling water over and over again, and there is a distinct economy in so doing when the water must be paid for. The usual arrangement (Fig. 41) consists of a vertical galvanized iron water tank of considerable capacity connected at its bottom to the lower part of the jacket, and near its top to the upper part of the jacket. The water in the jacket being heated, rises and flows to the upper part of the tank, where it cools by contact with the air and with the sides of the tank. Cold water from the bottom of the tank flows to the cylinder jacket to take its place. A continuous circulation is maintained by the difference of density between the cold and the heated water. In large engines when a large amount of water must be circulated, this method is generally too cumbrous and the water is taken from some constant source of supply, such as the city mains. The piping and valves are always so arranged that it is possible to draw the water from the jacket.

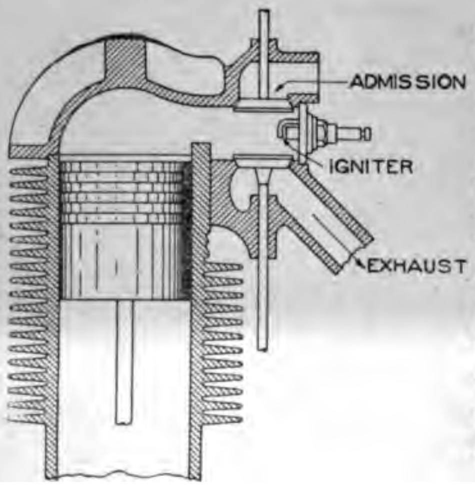

Fig. 40. Arrangement of Cylinder for Air Cooling.

The Explosive Mixture. The air used in the engine may be taken from the engine room or from the outside. The inrush of air to the air pipe makes a noise which is often objectionable in the engine room, but which can be greatly reduced if the air is taken from a large chamber, as in Fig. 10, where it is taken from the base of the engine.

If the gas is taken from the city mains, the intermittent action of the engine in admitting gas will cause considerable fluctuation of pressure in the supply pipe, which is not only undesirable in that it makes variable the amount of gas admitted, but also causes flickering of any lights supplied from the same pipe. To reduce this fluctuation it is usual to insert in the gas supply pipe a rubber bag (see Fig. 11), which collapses partly during the admission stroke and fills out again during the other strokes. Any enlargement in the gas supply pipe will serve the same purpose, but the flexible rubber bag is more effective than a mere enlargement.

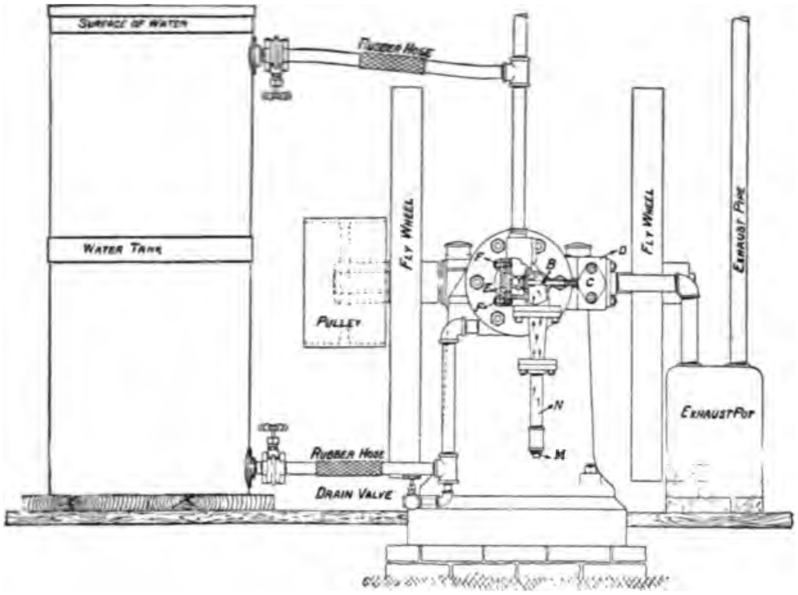

Fig. 41. Arrangement of Water Circulation for Jacket.

The air and gas should be mixed as thoroughly as possible on their way to the cylinder. This is satisfactorily accomplished if the air and gas have to pass through a common admission valve after they are mixed, as in Figs. 8, 10 and 33. The strength of the mixture is adjusted by throttling the gas supply, the air supply being left uncontrolled.

The Exhaust, if allowed to escape direct from an exhaust pipe of uniform cross section, is generally a source of annoyance by reason of the loud noise which it makes. This noise is greatly

e pipe discharges into an exhaust chamber or pot (see
ore going to the air. The injection of water into the
is also useful in reducing the noise in large engines.
noise nearly imperceptible a good plan is to have the
ge near the bottom of a pit filled with large stones.

ations of the Otto Cycle. Although the gas engine
to cycle will give a higher efficiency than any steam
is nevertheless desirable to increase its efficiency as much
ssible. Its efficiency has been shown to depend on the amount of compression, and the obvious way of increasing the efficiency is to decrease the clearance and thereby increase the compression pressure. The amount of compression that can be used is limited by two considerations. The first is that it is not commercially practicable to construct engines which will work properly under very high pressures rapidly imposed by explosions. With an engine compressing the charge to 100 lb. pressure and using a strong explosive mixture, the pressure in the cylinder rises suddenly to about 350 lb., and this is at present about the practicable limit. If the explosive mixture is weak, the compression may be increased; with very weak mixtures a compression to 200 lb. is sometimes used, and results in a maximum pressure of about 300 lb.

The second objection to the use of high compression is that the rise in temperature of the mixture resulting from the compression may easily be sufficient to explode the mixture before the piston has reached the end of its stroke. Such pre-ignition of the charge, tending to force the piston back, gives rise to a great shock and is very destructive to the engine, besides reducing its efficiency, and is consequently to be avoided. Pre-ignition may occur even with low compression if any part of the clearance is not water-jacketed, or if there is any metallic projection into the clearance space. Such unjacketed parts, or projections, not being properly cooled, are liable to be raised to a temperature high enough to cause the ignition of the charge. This often forces water-jacketing of the exhaust valve and of the piston in engines of large size.

Another method of increasing the efficiency is by what is known as *scavenging* the cylinder. In the ordinary Otto cycle the charge which is compressed consists of a mixture of fresh air and gas with the burned gases remaining in the clearance space from the previous cycle. If these burned gases are expelled from the cylinder by a charge of fresh air before the admission of the explosive charge, the force of the explosion and the efficiency are increased. The clearing out or scavenging of the cylinder with fresh air has been accomplished in several ways. The simplest method is by the use of an exhaust pipe of such length that the gases exhausting from the cycle with great velocity create a vacuum in the cylinder near the end of the exhaust stroke. This vacuum causes the automatic air admission valve to open and the consequent rush of air from the air valve to the exhaust port flushes out the cylinder, especially if the air and exhaust valves are on opposite sides of the clearance space. Occasional scavenging is obtained in engines governing on the hit-and-miss principle, each idle cycle flushing out the cylinder with the result that the succeeding explosion is of greater force than the normal explosion.

It has been pointed out already that the pressure at the end of the expansion in the Otto cycle is high, and that the efficiency of the cycle can be increased considerably if the gas is expanded more completely. Ordinary steam engine practice suggests that the more complete expansion can be obtained by *compounding*, but attempts so far to make a satisfactory compound gas engine have not proved very successful. The practical method of obtaining more complete expansion is to take into the cylinder a diminished charge. The two methods of accomplishing this have been discussed already. The only fundamental difference between engines using these two methods is that in one case the governor controls the amount of the opening of the admission valve, while in the other case it determines the instant at which the admission valve shall close.

One of the main objections urged against the Otto cycle is that it requires two revolutions of the engine for its completion, so that the expansion or motive stroke comes but once in four strokes. There results from this a very irregular driving effort, making large fly wheels necessary if the main shaft is to rotate uniformly, or else requiring the use of several engines working on the same shaft. The motive efforts can be made twice as frequent

if the cylinder is double acting, with admissions and explosions occurring on both sides of the piston. Many large engines are now being made double acting, but the practical troubles in keeping the piston, piston rod; cylinder and stuffing box cool enough for satisfactory working have prevented the use of double-acting cylinders in engines of small size.

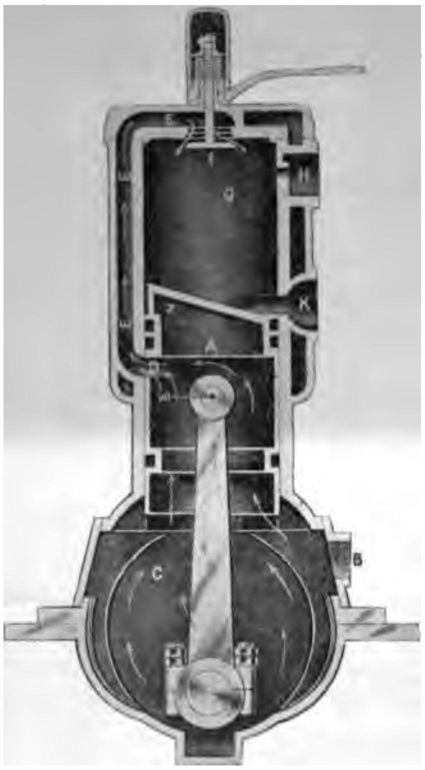

Fig. 42. Smalley Two-Cycle Engine.

An increased frequency of the expansion or motive stroke can be obtained by a slight modification of the Otto cycle which results in the cycle being completed in two strokes, and which is consequently called the *two-cycle* method. Engines using the two-cycle method give an impulse every revolution, and consequently not only give greater uniformity of speed of rotation of the crank shaft, but also develop nearly twice as much power as *four-cycle* or Otto cycle engines of the same size. Moreover, they are generally of great simplicity, having fewer valves than the four-cycle engines. An example is shown in Figs. 42 and 43 of a two-cycle engine of small size; Fig. 42 is a vertical section showing the piston at the bottom of its stroke, and Fig. 43 is a vertical section in a plane at right angles to the previous section plane and showing the piston at the top of its stroke. As the trunk piston A makes its upward stroke, it creates a partial vacuum below it in the closed crank chamber C, and draws in the explosive charge through B. On the downward stroke the charge below the piston is compressed to about 10 lb. pressure in the crank chamber C, the admission through B being controlled by an automatic valve which closes when the pressure in C exceeds the atmospheric pressure. When the piston reaches the lower end of its stroke, it uncovers the exhaust port K and at the same time brings the admission port D in the piston opposite the by-pass opening E E E, and permits the compressed charge to enter the cylinder G through the automatic admission valve *f* as soon as the pressure in the cylinder falls below that of the compressed charge. The return of the piston shuts off the admission through E and the exhaust through K and compresses the charge into the clearance space. The charge is then exploded (Fig. 43) and the piston makes its down or motive stroke. Near the end of the down stroke, after the opening of the exhaust port K, the admission of the charge at the top of the cylinder sweeps the burned gases out, the complete escape being facilitated by the oblique form (Fig. 42) of the top of the piston. The engine is so designed that the piston on its return stroke covers the exhaust port K just in time to prevent the escape of any of the entering charge. The processes described above and below the piston are simultaneous, the upstroke being accompanied by the admission below the piston and compression above it, while the down stroke has expansion above the piston and a slight compression below it. The very short interval of time between the beginning of the exhaust and the admission of the new charge (which enters as soon as the pressure in the cylinder has fallen enough to permit the admission valve to open), makes premature ignition of the charge, or *back firing*, of not infrequent occurrence. If

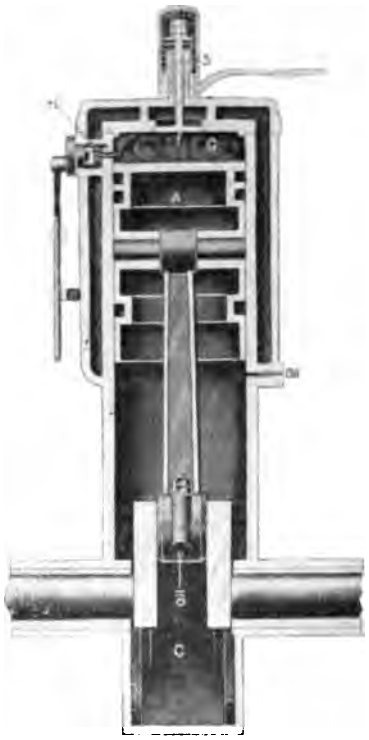

Fig. 43. Smalley Two-Cycle Engine.

the mixture is weak, or the speed is very high, so that the charge is still burning when the exhaust opens, or if the frequency of the explosions bring any part of the cylinder to a red heat, the charge will be ignited on entering, and the explosion then travels back through E E E to the crank case, which has to be made strong enough to resist it. In large engines the charge is compressed by a separate pump and not in the crank case. A modification of this engine makes the construction even more simple, so that the only valve on the engine is the automatic valve admitting the charge to the crank case. In this engine (Fig. 44) the series of operations is precisely similar to that just described. The only difference is in the by-pass connection E, which has no valve between it and the cylinder. The exhaust is made to open a little earlier than the admission, so as to make sure that the pressure in the cylinder shall have fallen below the pressure of the slightly compressed charge when the admission port opens. If the opening of the exhaust and admission ports were simultaneous, as in the engine just described, some of the exhaust gases would force their way through E to the crank case, igniting the charge there. The piston is so shaped that the entering charge is directed to the top of the cylinder, forcing out the burned gases before any of the charge can escape through the exhaust port.

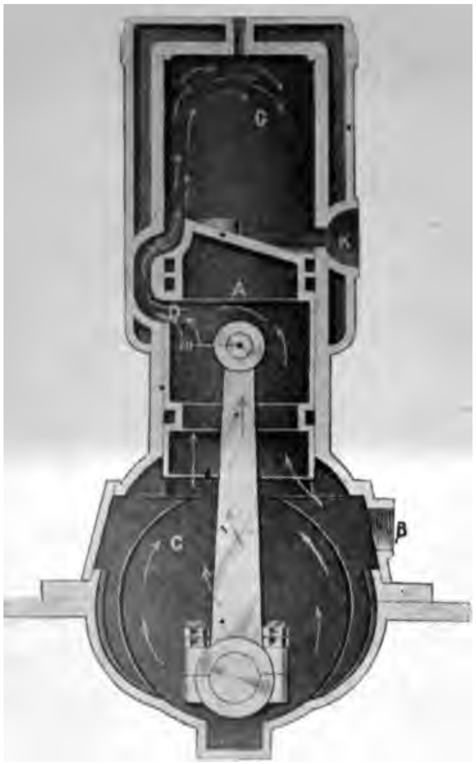

Fig. 44. Smalley Two-Cycle Engine.

The fact that the exhaust port is open while admission is taking place makes it always possible in a two-cycle engine that some of the charge may be lost through the exhaust. If the exhaust closes early, so as to diminish the probability of such loss, it will cause the retention in the cylinder of an unnecessarily large volume of burned gases, with consequent decrease in power and efficiency. The two-cycle engine may then be regarded as a modification valuable from the point of view of the more uniform turning movement, compactness and simplicity, but it is always likely to be inferior in efficiency. In some large engines the air and the gas are compressed separately; air alone is admitted at first, expelling the exhaust gases, and then the gas valve opens, admitting gas with the air. In this way the cylinder can be thoroughly scavenged and the exhaust be closed before any of the explosive charge reaches it.

Gas Engine Fuels. The fuels used in gas engines are very variable in origin, in composition, and in heat value. They consist almost entirely of various compounds of the chemical elements of carbon, hydrogen and oxygen, diluted with more or less nitrogen. In those regions where *natural gas* occurs, that fuel is used almost exclusively in the gas engine; but in most regions the gas has to be made either from solid or from liquid fluids. The use of liquid fuels will be considered later in connection with the discussion of the oil engine. In most towns of moderate size there is available *illuminating gas* made from coal. The illuminating gas is made by one of two processes giving either *coal gas* or *water gas*.

Coal Gas is made by heating the coal in a retort away from contact with the air, so that no combustion takes place. The hydrocarbon gases in the coal are driven off by the heat, and after undergoing various purifying processes, are collected in a holder. The non-volatile part of the coal remains as coke. The gas consists mainly of hydrocarbons and has a high heating value.

Water Gas is made from a non-gaseous fuel such as anthracite coal or coke, by an intermittent process. Air is blown through a bed of coal several feet thick until the coal is incandescent, the products of combustion being permitted to escape. Then a jet of steam is blown through the incandescent fuel, and is thereby broken up into its constituent elements—hydrogen and oxygen.

The oxygen combines with the carbon of the fuel to form carbon monoxide, CO; the hydrogen goes off unchanged. The passage of the steam quickly cools the coal, and air has to be blown through again. The only gas collected is that generated during the steam blow; it consists principally of hydrogen and carbon monoxide, and has a much lower heating value per cubic foot than coal gas. The whole of the coal is consumed in this process.

Both coal gas and water gas are excellent fuels for use in a gas engine, but as they have gone through certain processes for cleansing them and increasing their illuminating power, which increase the cost of the gas but do not add materially to its value for gas engine use, and since also the cost to the consumer is considerably greater than the cost of production, they are not economical fuels. Such fuels should be used only when the engine is very small or its operation very infrequent.

For engines of 50 H.P. or over, which are in regular operation, it is practically always more economical to generate the gas in a special *gas producer* than to use illuminating gas. In the gas producer either air alone, or generally both air and steam, are sent through a thick bed of coal. The oxygen of the air on first striking the zone of the incandescent coal combines with the carbon to form carbon dioxide, CO_2, but this on passing through the burning coal above, is reduced to carbon monoxide, CO, which escapes with the hydrogen and carbon monoxide resulting from the action of the steam on red hot coal, and with the nitrogen which came in with the air. The resulting gas therefore consists almost entirely of carbon monoxide, hydrogen and nitrogen. The large amount of nitrogen in the air (79 volumes in 100) makes the producer gas contain fifty per cent or more of that inert gas, and consequently gives it a low heat value.

A good example of a gas producer is shown in Fig. 45 under working conditions. The bed of coal, several feet thick, rests on a bed of ashes of about equal thickness, the ashes being supported on a solid circular table *a*. The blast pipe *b* terminates near the top of the bed of ashes, the blast being discharged radially so as not to concentrate the combustion. The blast is generally produced by a steam jet blower, but sometimes a fan blower is used. In the latter case steam is mixed with the air in the blast pipe so as to keep down the temperature of the producer and to soften any clinkers that form. Fresh coal is supplied by a continuous automatic feeding device on top of the producer, which spreads the coal in a uniform layer over the upper surface. In many producers the coal is merely dumped in from above at intervals, and has to be spread by hand. The intermittent charging has the disadvantage that it causes considerable variation in the condition of the fire, and consequently in the composition of the gas generated. The bed of ashes is maintained of the desired depth and the surplus ashes removed by rotating the grate *a* by means of gears worked through the crank *c*. As the grate is placed at some distance below the conical casing or *bosh*, the ash discharges uniformly around its periphery when it is revolved. This causes a uniform settling of the bed of ash, and also lets the bed of fuel settle so as to close up any channels in it which have been formed by the blast. The scrapers *d*, projecting a short distance into the ash bed, help the discharge of ash from the grate. The depth of the bed of ashes ensures that the ash is completely burned and cooled before it is finally discharged.

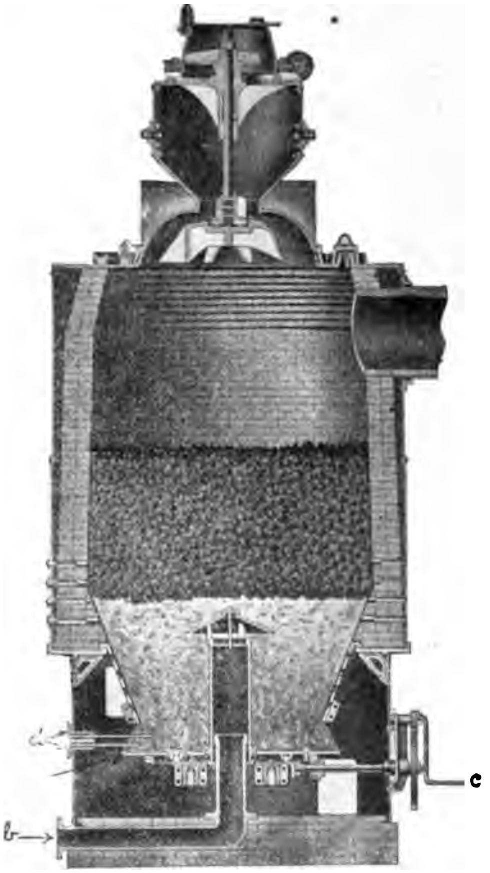

Fig. 45. Taylor Gas Producer.

Producer Plants are of two kinds, according as the flow of air through the producer is caused by air being forced in from below or by a partial vacuum being created above the fuel. The former is called a *pressure plant*, the latter a *suction plant*.

The general arrangement of the *pressure type* of producer gas plant is shown in Fig. 46, in which the arrows indicate the direction of flow of the gas. A small boiler supplies steam to the blower. The gas escapes from the producer at a high temperature and goes to an economizer, where it gives up much of its heat either to fresh air, which is about to be forced through the producer, or else to water, the vapor which mixes with the air. The gas then passes to the scrubber, where it meets a spray of cold water, which further cools it and takes from it dust and solid impurity, after which it goes to the purifier for the extraction by chemical process of certain undesirable components and for the completion of the removal of solids, and thence to the gas holder. If anthracite coal or coke are used, very little chemical purification is necessary; if bituminous coal is being burned, the cleaning is somewhat more complicated, as the tar and other troublesome substances in the gas have to be extracted before it can be used.

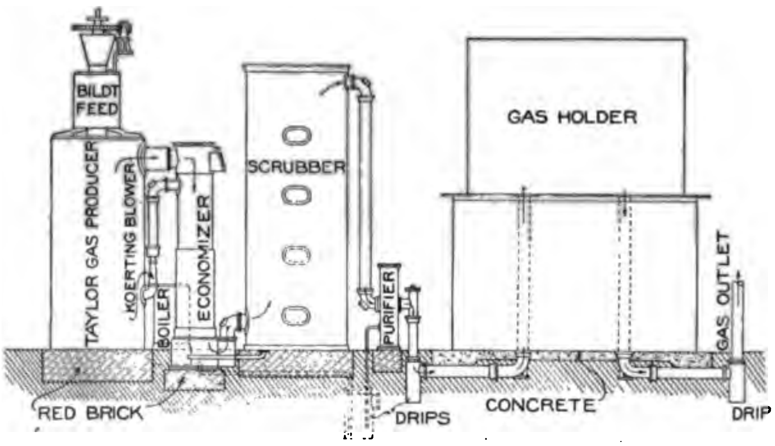

Fig. 46. Producer Plant—Pressure Type.

The *suction type* of gas producer plant can be used only when the operation of the engine is continuous for long periods. It has considerable advantage over the pressure type in compactness, but

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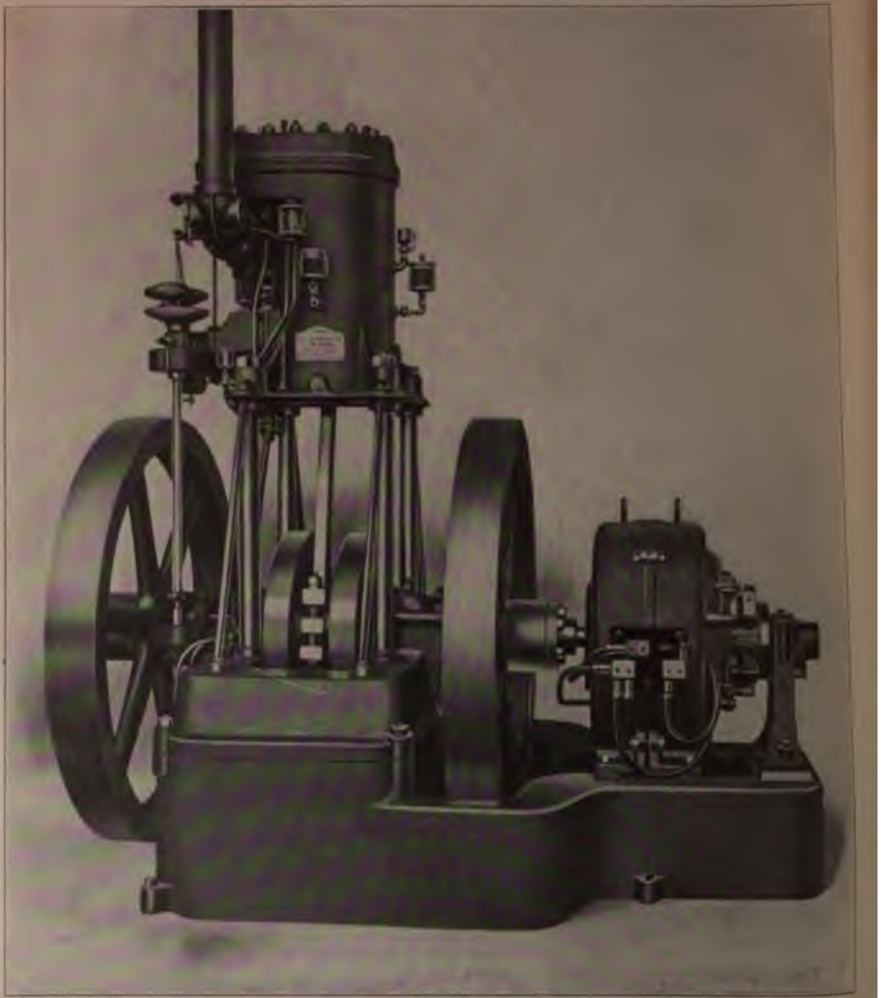

15 KILOWATT, 250 LIGHT.
Secor Oil-Electric Generating Plant.

is rather troublesome to start. The flow of air and vapor through the fuel in the producer or generator (Fig. 47) is dependent on the sucking action of the engine each time it takes in a charge, so that no boiler is needed to produce the blast. The volume of gas generated is always equal to the amount that the engine uses, so that no gas holder is required between the producer and the engine, its place being taken by a small gas tank. To start the producer working, a small hand or belt-driven blower is used, and the products of combustion are sent past a by-pass valve directly to the atmosphere until the escaping gas will burn steadily. The by-pass valve is then closed, and the gas is forced through the scrubber and purifier into the gas tank and the whole apparatus is filled with gas. When good gas appears at a test cock near the engine, the engine is put in operation and the blower is stopped, its function being performed thereafter by the engine. The hot gases escaping from the generator go first through an economizer or vaporizer (not shown in Fig. 47), and the steam formed there is conducted to the under side of the grate of the producer and is sucked through with the air.

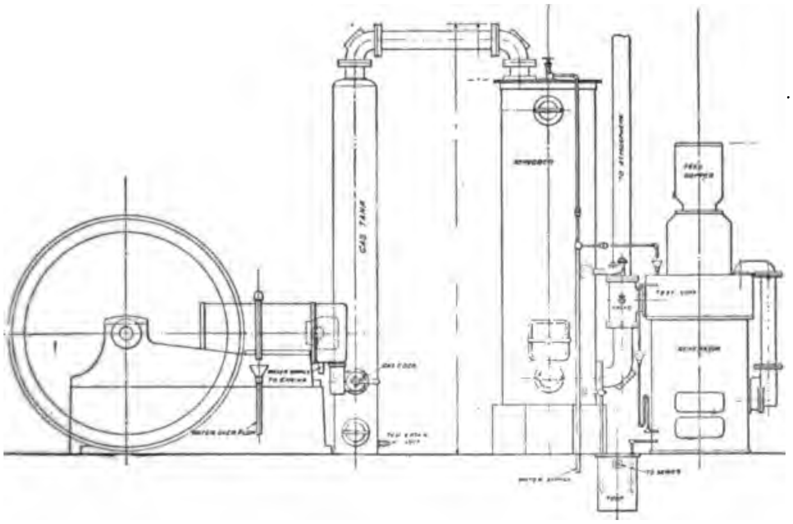

Fig. 47. Producer Plant—Suction Type.

Owing to the resistance offered by the fuel, scrubber, and other parts of the plant to the passage of the gas, its pressure on reaching the engine is considerably below the atmospheric pressure. This causes a decrease in the weight of the charge taken to the engine, and so makes the power of the engine less than when pressure gas is used. In order to get the high compression which is necessary to ensure ignition with a weak gas supplied at a low pressure, the clearance in the engine using suction gas is smaller than in other engines using the same cycle. It is not safe to use such an engine with illuminating gas, as the pressures resulting from explosion would be excessive. When in some cases illuminating gas is used to start the engine, a special device is used to exhaust some of the charge during the compression period, and so to reduce the compression pressure.

An efficient producer of either the pressure or suction type will waste not more than fifteen to twenty per cent. of the heat of combustion of the coal in converting it into gas—that is, the gas on burning will give up eighty to eighty-five per cent. of the heat of combustion of the coal. Its efficiency exceeds that of a steam boiler. If the gas produced is a weak one, it is produced in greater volume, and it has to be mixed with a much smaller volume of air than is required for illuminating gas. For example, ordinary coal gas must have at least six parts of air to one of gas, whereas producer gas requires a minimum of about one-and-a-quarter parts of air to one of gas.

The heat liberated by the combustion of a cubic foot of each of the gases discussed is as follows:

Natural gas.	900 – 1000	B.T.U.
Coal gas	650 – 700	"
Water gas	300	"
Producer gas.............	120 – 150	"

The power which can be developed in an engine does not depend upon the heat of combustion of a cubic foot of the fuel but on the heat of combustion of a cubic foot of the explosive mixture. The difference in the amounts of air necessary for combustion with the different gases makes the heats of combustion per cubic foot of the explosive mixture much more nearly equal than the heats of combustion per cubic foot of the fuel. Thus, when mixed with just sufficient air for complete combustion, natural gas, coal gas, and water gas will all give up about 90 B.T.U. per cu. ft. of the explosive mixture, while producer gas gives up about 65 B.T.U. An engine will consequently develop about the same power whether using natural gas, coal gas or water gas; with producer gas it will develop considerably less power.

Blast Furnace Gas. Besides the gases produced from solid fuels for illuminating or power purposes, there are waste gases escaping in many industrial processes (notably in the production of pig iron) which have considerable heat value. These gases have been utilized by burning them under a boiler for the generation of steam, but they are generally satisfactory, after cleansing, as fuels for a gas engine, and when so used give three or four times more power than when used for steam generation. The gases escaping from a blast furnace have a heat value of about 100 B.T.U. per cubic foot. Even when the gas is so poor that it will not burn under a boiler, it can be made to burn satisfactorily in a gas engine because of the high compression to which it is subjected.

The Care of a Gas Engine. For the successful operation of a gas engine intelligent care and accurate adjustment are necessary, and also an understanding of the processes going on in the cylinder. It sometimes happens that the engine fails to start, although the ordinary starting operations have been carried out faithfully. The most common causes of this difficulty are incorrect strength of mixture, failure of ignition and leakage of the charge. The setting of the gas valve which gives a satisfactory mixture one day may give a non-explosive mixture on the following day as a result of variation of pressure of the gas or other change. The strength of the mixture should be varied in case of failure to start. If this is ineffective, the ignition should be tried. The batteries may have run down as a result of much use or of short circuiting, and should be tested by short circuiting momentarily, when they should give a bright spark. Too strong a current is undesirable, as it burns the contact points rapidly. It is well to have on hand a spare set of cells for putting in circuit. There should always be a switch in the battery circuit, which should be thrown out when the engine is shut down, so as to prevent short circuiting. If the battery is in good condition, the trouble may be with the electrodes, either by their having become fouled or wet, or, in the make-and-break system, by a gumming of the spindle of the moving electrode which makes it sticky and slow in action. The igniter plug should be withdrawn and the electrodes examined. The whole igniter circuit should be examined for short circuits.

If the trouble is not with the igniter it may be caused by leakage of the charge. To test this, the engine, if not too large, is pulled over by hand. The resistance to turning on the compression stroke should be very considerable. If the resistance is not very great, or decreases, the compressed charge is escaping. The leakage may be either past the piston, the igniter plug or the valves. If the leakage is past the piston, it is either due to the wearing of the cylinder or to the sticking of the piston rings. The latter is very liable to occur after a while, especially if the cylinder has been permitted to get very hot, and can be remedied by taking the piston out and loosening and cleaning the rings with kerosene. A leakage past the valves is due either to gumming of the valves or to other deposit which keeps the valve off its seat, to wearing of the valve, or to sticking of the valve stem in its guide as a result of imperfect lubrication. The gumming and wear of the exhaust valve is the most common of the causes of leakage and may be remedied by grinding the valve on its seat with flour of emery and oil.

The presence of water in the cylinder, which has leaked in from the jacket through imperfect joints, sometimes causes the electrodes to become wet and prevents the engine starting. In some engines the possibility of this particular trouble is avoided by a special design of the jacket which has no joints communicating with the inside of the cylinder.

The cylinder oil that is used in steam engines cannot be used in gas engines, as it carbonizes at the high temperature of the explosion, and forms a deposit in the cylinder and on the exhaust valve. A much lighter oil is used, and even this if supplied in excess causes a gradual accumulation of hard deposit in the cylinder which must be cleared out occasionally. Apart from its interference with the action of the igniter and exhaust valve, it is liable to cause premature ignition by being raised to incandescence.

Cold water must be kept circulating through the jackets whenever the engine is running, being started as soon as the cylinder warms up. A stoppage of this flow, even for a comparatively short time, is liable to have a disastrous effect upon the cylinder. A gradual accumulation of sediment may occur in the water jacket, with a consequent reduction in its efficiency. On shutting down, it is always better to drain the jacket, which not only prevents the possibility of its freezing up in winter, but also tends to clear it of deposit of sediment. Generally, however, the jackets are drained only in cold weather.

In the running of a gas engine—especially under light loads—very loud and alarming explosions are sometimes heard in the admission pipe or in the exhaust pipe. The *back firing* in the admission pipe nearly always results from a leaky admission valve. The explosions in the exhaust, indicating as they do the presence of explosive gases in the exhaust pipe, are caused either by the use of a mixture which is too weak or by faulty ignition. If the mixture is too weak, the charge taken in just after an explosion may fail to ignite because it is mixed with the products of the previous explosion, while the next charge taken in may explode because it does not mix with burned gas but with the weak charge in the clearance. The hot exhaust gases ignite the weak mixture which was rejected unburned to the exhaust at the previous cycle. If the ignition is imperfect, a good mixture may fail to explode and be exhausted, and then ignited in the exhaust pipe by the next exhaust of hot gases.

Large Gas Engines. A very rapid development has taken place in the size to which gas engines are built until they are now made as powerful as the largest steam engines. To obtain large powers and to get the desired uniformity of crank shaft movement, multi-cylinders and multi-cranks are commonly used. When double-acting cylinders are used, the piston being subjected to high temperatures on both sides, becomes too hot unless there is a circulation of cooling water within it. If the piston is to be water-cooled, the piston rod is made hollow and is furnished with an internal tube. A water pipe is attached to the piston rod near the crosshead, by means of swing joints, and a current of water flows through the internal tube to the piston, circulates through it in a regular path, returns along the annular space around the internal tube in the hollow rod and escapes through other pipes with swing joints.

To obtain an impulse each stroke from a single cylinder engine (Fig. 48) a two-cycle double-acting engine may be used. In this engine the exhaust occurs by the uncovering of the ports in the middle of the cylinder when the engine is near the end of its stroke. The air and gas that are to be admitted are compressed separately in the two cylinders seen in the lower part of the sectional plan. When in consequence of the opening of the exhaust, the pressure in the cylinder falls below that to which the air is compressed, air enters through an automatic valve (not shown) at the top of the cylinder, and rushing toward the exhaust port, sweeps the cylinder clear of burned gas. The gas admission valve opens immediately afterwards, and an explosive charge enters; but before this can get to the exhaust port, the return of the piston stops the exhaust. In this way the cylinder is kept clear of burned gases without the loss of any of the entering charge. The propagation of the explosion in the cylinders of large engines is comparatively slow, so that two or more igniters are sometimes used so as to start the explosion in several places simultaneously.

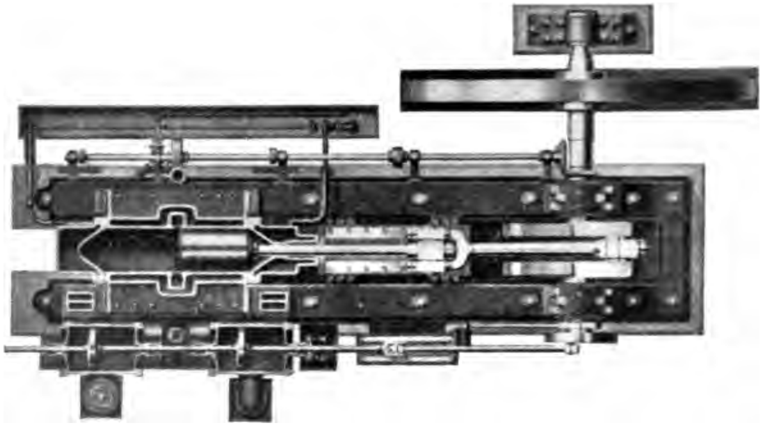

Fig. 48. Koerting Gas Engine.

Fuel Consumption. The consumption of fuel in a gas engine running at its rated load when natural gas is used, is from 13 to

17 cu. ft. per B.H.P. per hour; with coal gas 15 to 19 cu. ft. per B.H.P. per hour, and with producer gas about 1¼ lb. of coal per B.H.P. per hour. These are average good results; large engines show higher economy than smaller engines and have given a B.H.P. with a consumption of 1 lb. of coal per hour.

Oil Engines. The fuel used in oil engines is generally *crude petroleum* or some of the oils derived from it by the process of refining. Crude petroleum is not a simple chemical substance, but is a mixture of a very large number of different hydrocarbons having widely varying properties. If crude petroleum is slowly heated, it gives off as vapor its various constituent elements, the more volatile being given off at the lower temperatures, and the residue becoming continuously more dense and more viscous. In the refining of petroleum the vapors given off at various temperatures are condensed and collected separately, and the names given to the various products are an index chiefly to the temperature at which they give off their vapors. The most volatile of the ordinary products contains all the elements that vaporize at a temperature below 160° F., and is called *gasoline*. It gives off some of its lighter vapors at the ordinary temperature of the air, and as these vapors are highly combustible, gasoline is quite dangerous. When mixed with from eight to twenty parts of air, it forms an explosive mixture, which gives a more rapid explosion and consequently higher pressure than mixtures of equal heat value used in the gas engine. When exposed to the air, the lighter vapors escape, leaving behind a heavier and less volatile oil.

If petroleum which has been heated for some time at 160° is slowly raised in temperature to 250° F., a new and heavier series of vapors will be given off, which, when condensed and collected, are called *benzine* or *naphtha*. On still further raising the temperature from 250° F. to 350° F., a still heavier series of vapors is given off, forming the oil known as *kerosene*. Kerosene will not give off inflammable vapors till it is heated to about 120° F., so that it is comparatively safe and also will not change or deteriorate when stored under ordinary conditions. It is more difficult to burn satisfactorily than is gasoline, and when subjected to a high temperature with insufficient air for its combustion, it decomposes and deposits its carbon as a hard cake on the walls of the containing vessel. The dense petroleum which remains after the kerosene has been driven off is called *fuel oil.* If the fuel oil is subjected to still higher temperatures other and denser vapors are driven off, giving when collected *lubricating oils*, *cylinder oil* and *paraffine wax*, leaving finally a dense sticky mass which is known as *residuum.* The various oils can be distinguished in a general way by their densities, that is, their weights as compared with the weight of an equal volume of water. The density of gasoline is about .65, of kerosene about .80, of fuel oil about .82, of lubricating oils up to .92. If it is stated that an engine uses .76 kerosene, it means that the density of the kerosene is $\frac{76}{100}$ of that of water.

Crude petroleum, gasoline, kerosene and fuel oil are all used in oil engines. The cycle of operations through which the engine goes and the general structure of the engine may be the same for all these oils as for the gas engines already discussed; the only essential difference is in the addition of devices for supplying the oil to the cylinder and for its preparatory treatment.

Oil engines may be divided into two classes:

1. Those that convert the oil into a fine spray or a vapor before admitting it to the cylinder.
2. Those that deliver the oil to the cylinder in the liquid form.

Gasoline Engines. When gasoline is used, the vaporization is brought about by passing a current of air over or through the oil; the air escapes enriched with vapor of gasoline, and is said to be *carbureted.* The vessel in which the process takes place is called a *carburetor.* The carbureted air is too rich in fuel to be explosive, so that a further addition of air is necessary at the cylinder. The vaporization of part of the oil results in a lowering of temperature of the main body of the oil, and this reduces its volatility. In order to carry out the process satisfactorily with a uniform quality of the carbureted air, it is customary to heat the carburetor. This may be effected by making use of the heat either of the exhaust gases or of the escaping jacket water. The latter is more common with gasoline. A further advantage of moderate heating of the carburetor is that the denser constituents of the gasoline then become more volatile, so that the passage of the air through the oil does not result in depriving it of its lighter constituents by leaving a residue too dense to be used.

The carburetor shown in Fig. 49 consists of a vertical cylinder surrounded by a water jacket. Gasoline is pumped to the top of the cylinder and, passing through a spraying device, falls in a finely divided state. It meets an upward current of air drawn in through inlets at the bottom by the suction of the engine cylinder. The carbureted air goes from the top of the carburetor to the engine, while any unvaporized gasoline drains to the suction side of the gasoline pump and is returned later to the carburetor. The water jacket has circulating through it some of the heated jacket water from the cylinder. The actual temperature of the jacket is controlled by a thermostat which varies the amount of water circulating.

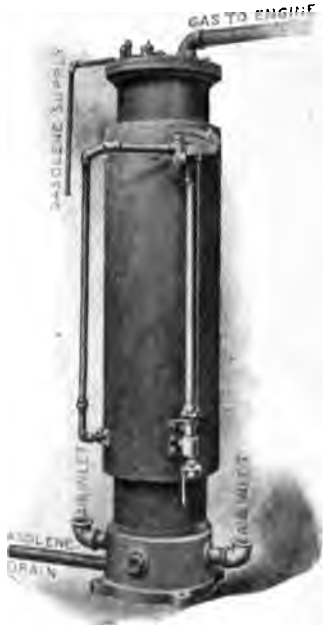

Fig. 49. Westinghouse Carburetor.

Gasoline is very fluid and atomizes completely when injected into a pipe through which a current of air is passing. The air in that case carries the gasoline with it, partly in the form of a mist and partly vaporized. This process is largely used in gasoline engines, and is illustrated in Fig. 50, which shows the whole arrangement of a gasoline plant. The gasoline tank is buried below the floor level and outside the building, so as to reduce the danger in case of fire or explosion, and also so that there can be no leakage of gasoline from the pipes when the engine is not running. The gasoline is taken through a strainer near the bottom of the tank and through the suction pipe by the action of a gasoline pump, which is worked from the cam shaft (see Fig. 11). It is then forced through the control valve A and is sprayed into the air pipe N through the jet H whenever the fuel admission valve G opens. A vertical branch of the discharge pipe from the gasoline pump has an overflow connecting with the tank. The pump always delivers more gasoline than is required, the excess being returned to the tank through the

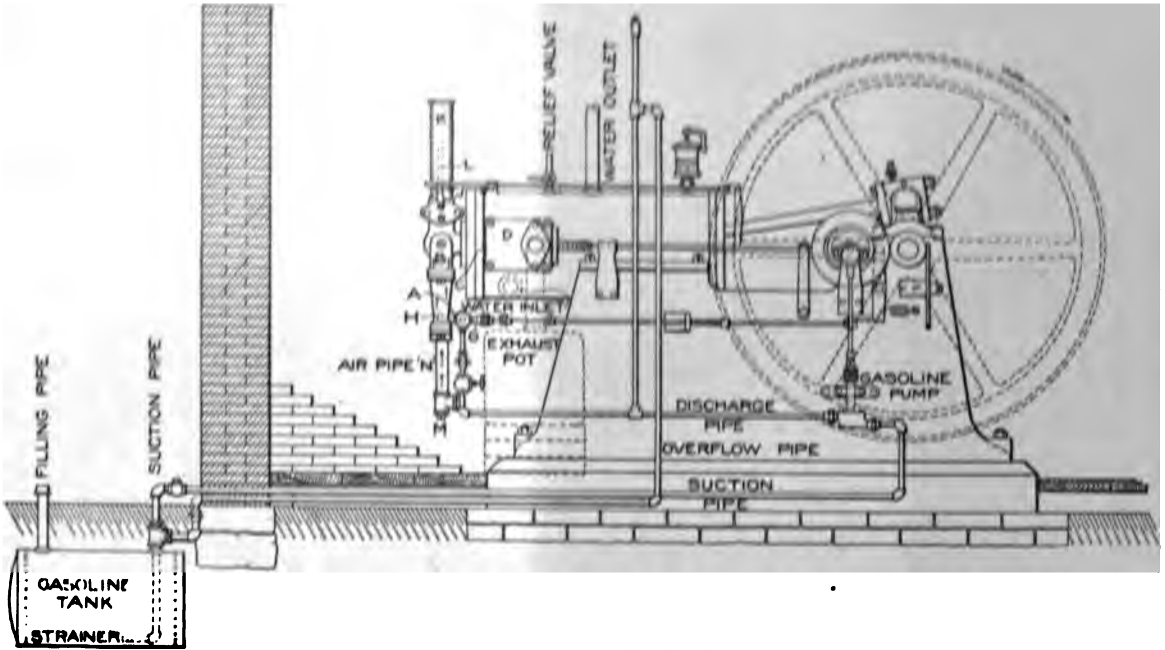

Fig. 50. Charter Gasoline Engine.

overflow pipe. This maintains a constant pressure of the gasoline depending only on the constant overflow level. With a given opening of the control valve A and a constant head on the gasoline, the amount of gasoline admitted each time remains constant.

An engine using gasoline is usually provided with a clearance space somewhat larger than that used in a gas engine, so as to prevent excessive pressures during explosion. The indicator card, Fig. 51, shows the rapidity and force of the explosion in a gasoline engine.

Kerosene and Crude Oil Engines. When kerosene or heavier oils are to be burned, different methods from those described for gasoline must be used. The kerosene is generally broken up in the carburetor into a fine spray or mist by a current of air; it is then sent to a vaporizer before being admitted to the cylinder. In the vaporizer the carbureted air is raised to a high temperature, the heat of the exhaust gases being utilized for this purpose, and the kerosene is converted into a vapor. Unless the kerosene is completely vaporized before admission to the cylinder, it is difficult to ensure its complete combustion. Some of the liquid kerosene in the cylinder *cracks* or breaks up into its elements as a result of the very high temperature to which it is subjected, and the carbon deposits itself on the piston and walls of the clearance space as a hard coating. The temperature in the vaporizer is not sufficient to crack the oil.

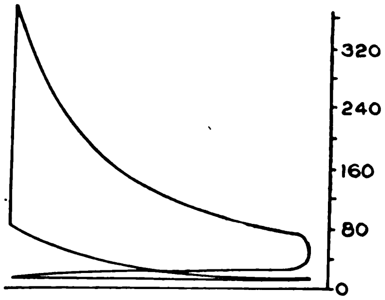

Fig. 51. Indicator Card of Gasoline Engine.

In some cases the carburetor or spraying device is omitted and the oil is pumped directly into a vaporizer or generator which converts the liquid into a vapor. A device of this kind for burning heavy crude oil, such as is found in the Texas oil fields, is shown in Fig. 52. The generator G is placed close to the engine so that the hot exhaust gases coming through the pipe N shall not be cooled before reaching it. The oil pumped by the engine goes through the pipe F to the small reservoir R on top of the generator, any excess returning by the overflow pipe O to the main supply tank. The amount of oil entering the generator is controlled by the throttle valve T at the reservoir R. The oil trickles down over surfaces which are heated by the exhaust gases and is partly or completely vaporized. Any unconsumed residue drains off through the cock D at the bottom of the generator. The temperature in the generator is regulated by a heat valve E, which may be set so as to circulate all or any part of the exhaust gases through the heating coil of the generator, the rest being sent directly to the exhaust pipe X. Air is drawn into the lower part of the generator through the pipe C, and the mixture of air and vapor leaving the top of the generator by the pipe B, meets a fresh supply of air arriving through the valve A before being admitted to the cylinder. When kerosene is to be used, the generator is very much smaller, but the general arrangement is similar.

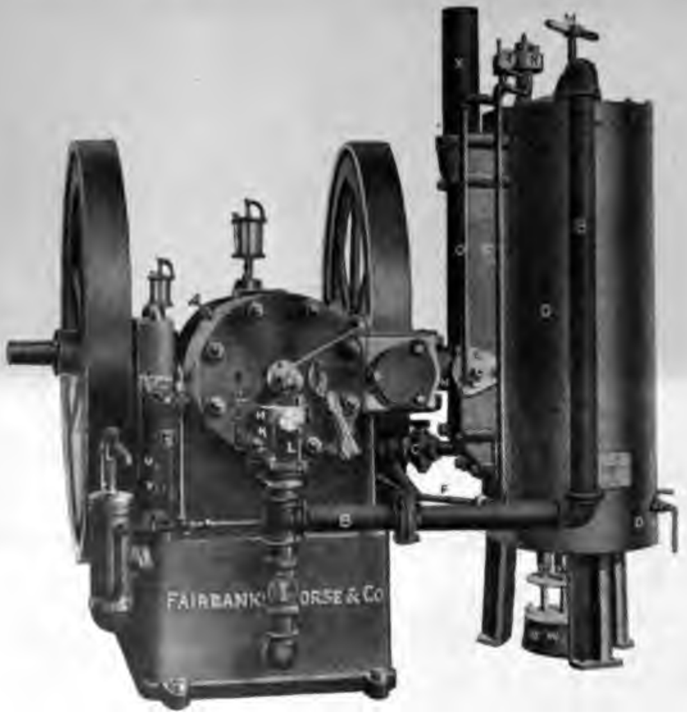

Fig. 52. Fairbanks-Morse Engine Arranged for Burning Heavy Crude Oil.

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Kerosene and the heavier oils can be used in oil engines without preliminary vaporization. One of the methods of accomplishing this is illustrated in Fig. 53, which is a longitudinal section of an engine using kerosene or crude oil. A combustion chamber or vaporizer A is attached to the end of the cylinder and communicates with it through a narrow neck B. The outer part of the vaporizer is unjacketed, and consequently is kept at a good red heat by the successive explosions. The engine follows the usual four stroke cycle. During the admission stroke air alone is admitted to the cylinder, while oil is injected into the combustion chamber and is vaporized there. During the return stroke the air is compressed into the vaporizer, mixes with the oil vapor and forms an explosive mixture which is ignited by the hot walls of the combustion chamber. The proportions of the combustion chamber are designed so that the explosion does not occur till near the end of the compression stroke. The fuel supply is regulated by the governor, which controls a by-pass permitting part of the discharge from the pump to return to the suction side. Before starting the engine the combustion chamber must be raised to a bright red heat by an external heater; but after starting, it is maintained in that condition by the explosions. The engine is of great simplicity since it dispenses with both igniter and mixing valve. The com-

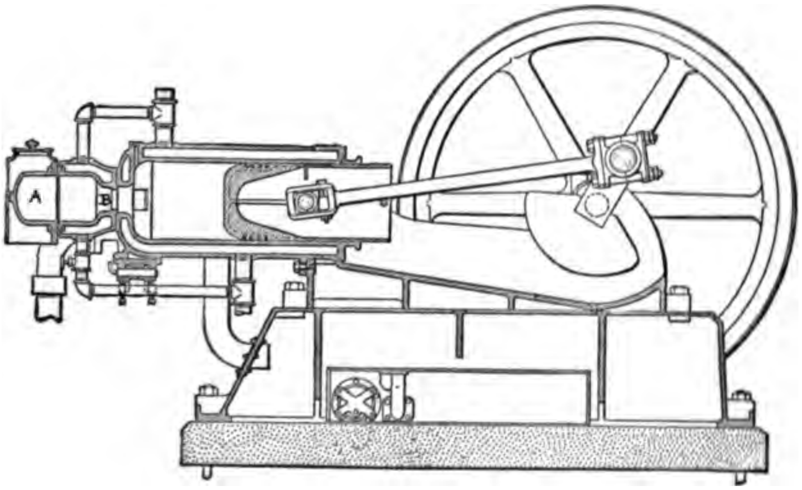

Fig. 53. Hornsby-Akroyd Oil Engine.

bustion chamber becomes coated with a deposit of carbon resulting from the cracking of the oil at the high temperature, but it is easily removed for cleaning.

The Diesel Cycle. All the engines discussed so far have operated on the Otto cycle or some modification of that cycle. There is coming into use for crude oil or fuel oil engines another cycle of operations, known as the Diesel cycle, which merits attention because of its high efficiency. The cycle will operate equally well with gas or gasoline, but is naturally used with the cheaper fuels. The Diesel cycle resembles the Otto cycle in requiring four strokes for its completion. The first out stroke draws into the cylinder a charge of pure air alone, without any admixture of the fuel. On the return stroke the air is compressed; and since the clearance in this engine is only about seven per cent of the cylinder volume, the pressure at the end of compression rises to about 500 lb. per sq. in., and the temperature of the air to about 1,000° F. As the high pressure is reached gradually, it does not cause a shock to the engine, such as an explosion to the same pressure would give. At the beginning of the second outstroke the oil admission valve opens and a charge of oil is blown into the cylinder in the form of a fine spray by a small quantity of air which has been compressed by a special compressor to about 550 lb. The moment the entering oil meets the highly heated air in the clearance space, it ignites and burns. The combustion goes on so long as the fuel is being blown in, usually for about one-tenth of the forward stroke; and since there is no large quantity burning at any instant, there is nothing in the nature of an explosion. Usually the heat generated by the combustion is not sufficient to prevent the pressure in the cylinder falling while the admission is taking place, so that the admission line on the indicator card falls below the constant pressure line as seen in the indicator card, Fig. 54. The method of burning is really essentially similar to that of an ordinary gas burner, and not to that of an explosive mixture, and consequently the oil will burn with any

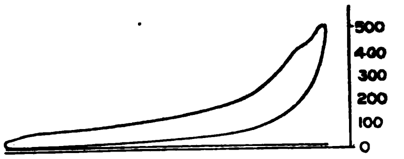

Fig. 54. Indicator Card--Diesel Engine.

excess of air present. After the admission valve has closed, the charge expands and then is exhausted on the return stroke. The indicator card, Fig. 54, shows the cycle of operations.

The general structure of the engine and a detail of the valves are shown in Figs. 55 and 56. The movement of the fuel admission valve is very slight, giving a narrow annular opening for the entry of the oil. Surrounding the valve spindle is a series of brass washers perforated parallel to the spindle by numerous small holes. The oil is pumped into the space around the valve spindle near its middle, and by capillary action finds its way between the washers and into the perforations. The air for fuel injection is

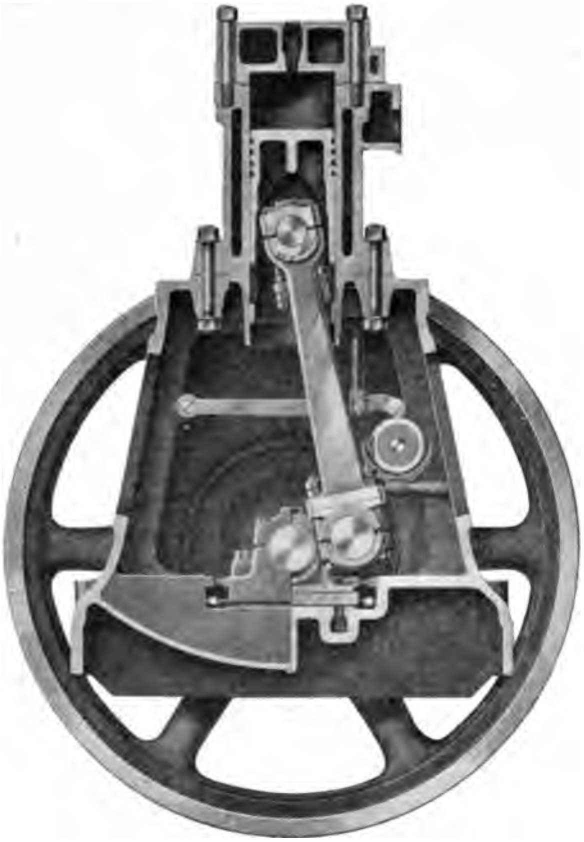

Fig. 55. Sectional View- Diesel Engine.

admitted through another pipe into the same space, but behind the oil, and because of its high pressure blows the oil into the cylinder when the valve opens. The amount of oil admitted is regulated by the governor which controls the time of opening of a by-pass connecting the discharge and suction sides of the oil pump. At light loads the oil is pumped to the fuel valve for part only of the admission period, and air above will enter past the valve for the rest of the period. The method of slow combustion in a large excess of highly heated air ensures very complete combustion even with the heaviest oils, so that there is no chance for the accumulation of a carbon deposit in the cylinder. The engine is started by compressed air from an auxiliary reservoir, a special starting valve being used for the purpose. Diesel engines have, under test, converted more than 35 per cent of the heat of combustion of the oil into work done in the cylinder.

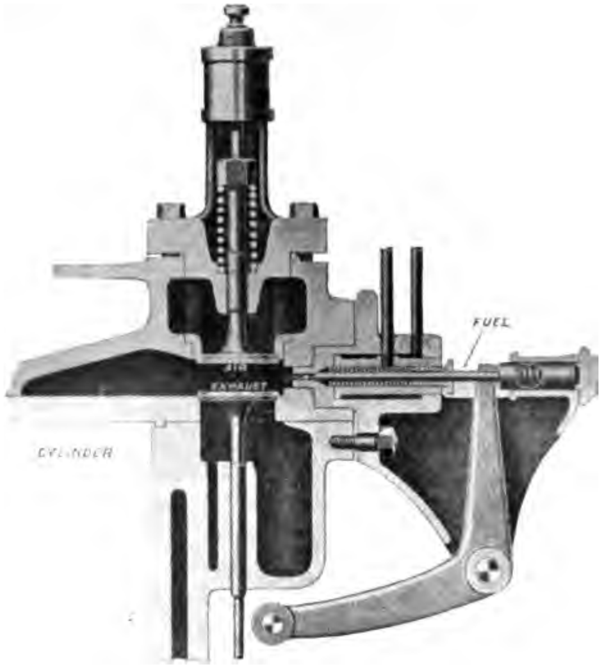

Fig. 56. Valves of Diesel Engine.

The Care of the Oil Engine. The same general precautions are necessary in running an oil engine as in running a gas engine, and the same troubles are liable to occur. The starting by hand of a gasoline engine of small size has been described already. If the engine fails to start, it will probably be because either too much or too little gasoline was admitted. The amount admitted for starting must be varied with the temperature of the cylinder. In cold weather about twice the normal amount must be used, while on the other hand, if the engine has been running and has been shut down for a short time only, a considerable diminished charge is necessary.

Great care must be taken by the use of suitable strainers that no solid foreign matter gets into the oil supply pipe, otherwise there is great liability to the obstruction of the flow. Owing to its more rapid explosion and to the greater richness of the explosive charge, a gasoline engine will develop more power than a gas engine of the same size, even when the latter uses natural gas.

Oil Consumption. The consumption of gasoline in an engine of small size averages about one-tenth of a gallon per brake horse power per hour. In the Diesel motor the average consumption of crude oil per brake horse power per hour is less than one-tenth of a gallon.

The field for the use of the oil engine is very extended. It is the most compact of the heat engines, requiring nothing equivalent to boiler or gas generator, and consequently is inherently the most suitable for purposes of transportation. Its extensive adoption for driving automobiles and motor boats is being followed by its application to locomotives and to large vessels. The absence of boiler and of heat generator losses makes it both potentially and actually the most efficient of all heat engines. The relative cost of power developed by oil, gas and steam engines depends on the cost of the oil and of coal, and this varies with the locality and the kind of oil or coal. In refining petroleum not more than ten per cent of the oil can be collected as gasoline, so that this oil, which is not the most easy to use, is not available in as large quantity, and consequently has a considerable higher cost than the heavier oils. Kerosene forms twenty-five to fifty per cent of the crude oil, and is consequently cheaper. Fuel oil and crude oil are the cheapest, but are also the most difficult to burn satisfactorily.

REVIEW QUESTIONS.

PRACTICAL TEST QUESTIONS.

In the foregoing sections of this Cyclopedia numerous illustrative examples are worked out in detail in order to show the application of the various methods and principles. Accompanying these are examples for practice which will aid the reader in fixing the principles in mind.

In the following pages are given a large number of test questions and problems which afford a valuable means of testing the reader's knowledge of the subjects treated. They will be found excellent practice for those preparing for College, Civil Service, or Engineer's License. In some cases numerical answers are given as a further aid in this work.

REVIEW QUESTIONS

ON THE SUBJECT OF

THE STEAM ENGINE.

PART I.

---

1. What should be the rim weight of a cast iron fly-wheel which is 15 feet in diameter, if the engine is of the Corliss type, with an 18″ × 36″ cylinder, and runs at 90 revolutions per minute?

2. In what way does the action of steam in the turbine engine differ from the action of steam in the ordinary reciprocating engine?

3. What is the theoretical height of a simple pendulum governor which makes 65 revolutions per minute?

4. In what way did Newcomen improve the steam engine?

5. Explain the difference between the condensing and the noncondensing engine, and show why the condenser increases the power of the engine.

6. Describe two methods of overcoming the danger due to high rotative speed in the steam turbine.

7. What two principles did James Watt follow in his experiments on the steam engine?

8. What types of engines have no fly-wheel? Why have they no fly-wheel?

9. What should be the thickness of rim for a cast iron fly-wheel weighing 4,480 pounds; the face is 12 inches wide, and the diameter of the wheel 15 feet?

10. Name some of Watt's inventions and improvements in the steam engine.

iat is meant by compound and triple expansion

iat is a Woolf engine? What is a tandem engine?
ound?

plain with sketch the action of a sight-feed cylinder

iat is the function of a governor?

iy is not the power given out by an engine constant
en time?

What are the relative advantages and disadvantages of turbine engines as compared with reciprocating engines?

17. What is the office of the fly-wheel?

18. How does the height of the governor affect its sensi-
ess?

19. Why are lubricants used? Name the requisites of good
ricants.

20. Name the advantages of the vertical engine.

21. Describe the two kinds of governors.

22. In what two ways do governors vary the work done on the piston?

23. How are high-speed engines oiled?

24. Which class of engine needs the larger fly-wheel, and why, high speed or low speed?

25. Describe the direct-acting steam pump.

26. How does the action of the Buckeye engine governor in changing the amount of steam admitted per stroke differ from the action of the straight-line engine governor?

27. What are the advantages and the principal disadvantages of the Corliss engine?

28. Name the requisites for high-speed reciprocating engines.

29. Why do pumping engines usually have devices for economy?

30. Explain how the expansion of steam is obtained in each of the two types of steam turbine.

31. If the height of a simple pendulum governor is 8 inches, what is the equivalent height if a weight equal to $1\frac{1}{2}$ times the weight of the balls is added?

REVIEW QUESTIONS

ON THE SUBJECT OF

STEAM ENGINE INDICATORS.

1. A Brumbo Pulley has the following dimensions: lever 80 inches long, stroke 32 inches and radius of sector 10 inches. What will be the length of the diagram?

2. Define Mean Effective Pressure.

3. When is a three-way cock used?

4. Describe Watt's Indicator.

5. A steam engine has a stroke of 30 inches, diameter of piston 18 inches, and the initial pressure 110 pounds (gage). If steam is admitted during the whole stroke and the exhaust is at atmospheric pressure, what is the theoretical M. E. P.?

6. In the above engine, what is the theoretical indicated horse-power if the engine is running at 75 revolutions per minute?
Ans. 318 I. H. P.

7. Why is a reducing motion necessary in taking indicator cards?

8. After taking an indicator card, how is the atmospheric line drawn?

9. Give the formula for finding the Indicated Horse-power, explain the meaning of the letters, and why is the denominator 33,000?

10. What is a good height and length for an indicator card? What data should be placed on the diagram?

11. Describe the process of taking a card.

12. Explain with sketch the method of finding the area of an indicator card by means of ordinates.

13. If you were told to find the I. H. P. of an engine what data would you require?

14. What peculiarity has the Tabor indicator?

15. What will be the fault with the card if the spring used in an indicator is too light? If too heavy?

16. If the I. H. P. is 85.6 and the efficiency 90 per cent, what is the available horse-power?

17. If the initial pressure of steam is 85 pounds absolute and the exhaust pressure 2 pounds above the vacuum, what is the thermal efficiency?

18. An indicator card gives a steam consumption of 28.6 pounds per I. H. P. per hour. If the mechanical efficiency is 87 per cent, what is the steam consumption per B. H. P.? Ans. 32.87 pounds.

19. A rope brake is attached to the fly wheel of an engine. The pulley is 36 inches in diameter and the rope 1 inch. If the spring balance registers 220 pounds and the weight is 20 pounds, what is the B. H. P. while the engine is running 300 revolutions per minute? Ans. 17.6 B. H. P.

20. Are the cards most reliable when taken by two indicators or when both are taken on the same paper by the same indicator?

21. What is the speed of an engine if the B. H. P. is 2.84, the length of arm of Prony brake 3 feet, and the weight at the end of the arm 35 pounds? Ans. 142 revolutions.

22. If the admission line of an indicator diagram curves backward, what does it indicate? How would you remedy it?

23. What is wire-drawing? Explain why it is undesirable.

24. The piston of an engine is 19 inches in diameter; the piston speed is 480 feet per minute. A card was taken with a 40-pound spring; the mean ordinate being 1.16 inches. What was the I. H. P.? Ans. 191.4 I. H. P.

25. What causes wavy lines in an indicator diagram?

26. What is the indicated horse-power of an engine 12″ × 18″ when running at 90 revolutions, the M. E. P. being 42 pounds?

REVIEW QUESTIONS

ON THE SUBJECT OF

VALVE GEARS.

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1. Why cannot the displacement of the piston be found in the same manner as that of the valve?

2. What is the throw of the eccentric if the valve travel is 3 inches and the rocker arm has the following dimensions: AB = 12 inches, AG = 15 inches. See Figs. 19 and 20.

3. Explain why the eccentric radius is not at right angles to the crank. What is lead?

4. Why are laps used? What effect does an increase of outside lap have upon steam distribution?

5. Describe an eccentric.

6. Sketch a plain slide valve and indicate ports, bridges, exhaust cavity, inside and outside lap.

7. Give the advantages of the Corliss valve gear.

8. With a connecting rod of ordinary length, explain why there is unequal steam distribution, if the valve is set for one end and if the laps are equal.

9. How do the following changes affect steam distribution with a plain slide valve. *a*. Increase in angular advance. *b*. Increase in eccentricity.

10. Describe the essential features of the Joy valve gear.

11. Referring to Fig. 16, will the line RO be perpendicular to XY when the piston is in the middle of the stroke?

12. What should be the diameter of the steam pipe for an 18″ × 24″ engine that makes 100 revolutions per minute?

Ans. 5 inches.

13. Describe how the governor regulates the engine by means of the shifting eccentric.

14. When setting the Meyer valve, is the main valve given a late or early cut-off? Why?

15. What relation between cut-off and compression can be obtained with the Meyer valve that is impossible with the ordinary slide valve?

16. Describe, with sketch, the Stephenson link.

17. Explain why an engine runs in the opposite direction if the valve is disconnected from one eccentric and connected to the opposite one.

18. What should be the width of the exhaust port?

19. What is the advantage of the piston valve?

20. What are the advantages of the radial valve gears?

21. What is one disadvantage of the Corliss gear?

22. How may a valve be most easily balanced to overcome weight of valve and gear as well as steam pressure?

23. How does the Gooch link differ from the Stephenson link?

24. Given in a certain valve:
Valve travel = 3 inches.
Lead angle = 6°.
Cut-off at $\frac{5}{8}$ stroke.
Compression at $\frac{3}{4}$ stroke.
Ratio of connecting rod to crank = 4.
Find by means of Zeuner's diagram:
Linear lead.
Outside lap.
Inside lap.
Angular advance.

25. What are "crossed rods"? How is steam distribution with crossed rods different from that with open rods?

26. How does the passage in the trick valve affect lead? How does it affect admission?

27. For what two reasons are link motions used on locomotives?

28. Why does bringing the block nearer the center of the Stephenson link cause cut-off to occur earlier?

29. What advantage has the double-ported slide valve?

30. What is meant by the mid-position of a valve?

REVIEW QUESTIONS

ON THE SUBJECT OF

THE STEAM ENGINE.

PART II.

---

1. Describe the method of finding the theoretical thermal efficiency.

2. Explain how there is any gain by using a steam jacket.

3. Explain the formula expressing the relation between pressure, volume and temperature of a perfect gas.

4. State Boyle's Law.

5. What effect does speed have on the economy of an engine?

6. Explain the advantages and disadvantages of superheating.

7. If steam exhausts into a jet condenser, how much water will be required (theoretically) to condense 10 pounds of steam, if the entering water is at 55° F, the discharge at 118° F and the vacuum gage reads 13.7?

8. At what position of the crank is the maximum turning moment?

9. If the feed water is heated, is the economy of the engine increased?

10. What is the absolute temperature of 150° F?

11. Other things being equal, is a large engine more economical than a small one? Why?

12. Explain initial condensation.

13. Describe the action of heat on a body of water under atmospheric pressure.

14. Define the terms: saturated steam, superheated steam, wet steam and dry steam.

15. Name the properties found in the steam tables.

16. How does a condenser increase the power of an engine?

17. Explain how re-evaporation is possible.

18. How is the amount of steam used per hour found?

19. Explain with sketch how the crank effort is more nearly constant if there is more than one crank on the shaft.

20. In what two ways does compounding effect a saving of steam?

21. Give the formula expressing approximately the relation between pressure and volume of steam. What is it called?

22. Find the probable I. H. P. of a high-speed triple-expansion engine for a battle-ship. The engine runs at 150 revolutions; total ratio of expansion 12; initial pressure 145 pounds (gage), condenser vacuum 14 inches of mercury; the low-pressure cylinder is 72″ × 36″.

Ans. 2567.2 I. H. P.

NOTE.— Use card factor of .60 and atmospheric pressure 14.7 pounds.

23. What is the temperature of saturated steam corresponding to a pressure of 120 pounds (gage)?

24. What is the heat equivalent of work? How many foot-pounds of work equal one B. T. U.?

25. How many B. T. U. are required to furnish energy sufficient to raise 500 pounds 5,280 feet? How many pounds of coal will furnish this heat, assuming 6,800 B. T. U. per pound are available?

$$\text{Ans.}\begin{cases}\text{3,393 B. T. U.}\\ \frac{1}{2}\text{ pound (about).}\end{cases}$$

26. How many cubic feet do 3 pounds of steam occupy when at 45 pounds absolute pressure?

27. How many B. T. U. are necessary to change 8 pounds of water at 320° F into steam at the corresponding temperature and pressure?

28. What is the actual thermal efficiency of an engine?

REVIEW QUESTIONS

ON THE SUBJECT OF

REFRIGERATION.

PART I.

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1. What is the principle upon which the modern method of refrigeration is based?

2. Why is it desirable to use a compressor?

3. A solution of chloride of sodium has a specific gravity of 1.1018. What is the freezing point in degrees Fahrenheit, and what is its density in degrees Salometer?

4. What do you consider the most important parts of a compressor?

5. Should the clearance of a compressor be small or great?

6. Why is ammonia so generally used in refrigeration?

7. What is the difference in the construction of a discharge valve and a suction valve?

8. Where should the oil separator be placed?

9. What is the difference between the direct-expansion system and the brine system?

10. What should be the displacement in the compressor for a seven-ton machine?

11. If a Fahrenheit thermometer in a room indicates a temperature of 38 degrees, what would be the reading of a Centigrade thermometer?

12. What temperature (Fahrenheit) would be indicated by a thermometer placed in an open vessel of liquid ammonia? What would be the temperature Reaumer?

13. Why is it important to have the spring of the discharge valve correctly adjusted?

14. What kind of valve is best adapted for regulating the flow of ammonia?

REVIEW QUESTIONS

ON THE SUBJECT OF

REFRIGERATION.

PART II.

---

1. What well-known apparatus used in steam engineering employs the rapid vaporization of water for producing cold?

2. At what temperature does ammonia boil if the pressure is 15 pounds per square inch?

3. In what way does refrigeration by carbonic anhydride gas differ from the ammonia process?

4. Describe Carré's first ice plant.

5. Why are oil and water traps used in the process of refrigeration by compressed air?

6. Why must the diameter of the steam cylinder be much greater than that of the gas cylinder in the carbonic anhydride compressor?

7. By what methods is the heat removed from compressed air so that it may be used as a refrigerating agent?

8. What is the working pressure in the ammonia still?

9. What is an average value for the working pressure of the carbonic anhydride process?

10. Why is the compressed-air system usually preferred to the ammonia process on board ship?

11. Discuss briefly the advantages and disadvantages of the slide valve and the poppet valve for air compressors.

12. State briefly the principle of the ammonia absorption process.

13. State briefly the cycle of operations in the process using carbonic anhydride gas.

14. Are gage glasses absolutely necessary to the successful working of an absorption system?

REVIEW QUESTIONS

ON THE SUBJECT OF

GAS AND OIL ENGINES.

---

1. Describe the cycle of operations in the Lenoir engine.

2. What is the usual pressure in the cylinder when the exhaust valve opens? What device is used to reduce the work necessary to raise a valve against that pressure?

3. What is an internal combustion motor, and what are its principal advantages over the external combustion motor?

4. Describe a gas producer plant and explain the differences between the pressure and the suction types of producer plant. Will a given gas engine develop as much power with producer gas as with coal gas?

5. What is the pressure at the end of compression in the ideal Otto cycle engine with 30% clearance?

6. What is the efficiency of the engine in the previous problem, and what is the probable efficiency of an actual engine with the same clearance?

7. What is an external combustion motor and what are its principal defects?

8. Describe the series of operations of the Otto cycle.

9. Explain the hit-and-miss system of governing gas engines. How is it carried out? What are its disadvantages?

10. What are the functions of the valves used in a gas engine? What kind of valve is generally used? Which of the valves may be automatic in action?

11. Draw and explain the indicator card of an engine using the Otto cycle with increased expansion.

12. Describe the procedure in starting a small gas engine by hand. How can a large gas engine be started?

13. What is the usual method of actuating the valves? What must be the speed of revolution of the side shaft?

14. Why is it possible (a) to compress to higher pressures, and (b) to use a larger excess of air with the Diesel cycle than with the Otto cycle? How is the governing carried out in the Diesel engine?

15. Describe the method of burning heavy oils used in the Hornsby-Akroyd engine.

16. Describe the procedure in starting an engine with gasoline. How much gasoline does an average engine use per brake horse power per hour?

17. What are the most common causes of failure to start in a gas engine? How would you set to work to find out the trouble?

18. How may kerosene be vaporized before being sent to the cylinder of the engine? What is meant by the cracking of kerosene, and how is it caused?

19. What is carbureted air? Describe the action of a carburetor for gasoline. Where should the gasoline tank be placed?

20. Draw the indicator card for the Otto cycle. What kind of compression occurs in this cycle?

21. What is back-firing, how is it caused and how remedied?

22. What is the average fuel consumption in a gas engine?

23. What are the fuels used in oil engines? How are they obtained and how may they be distinguished from one another?

24. Upon what does the efficiency of the Otto cycle depend? Under what conditions will the efficiency be high?

25. Describe hot tube ignition.

26. Explain the operation of a make-and-break igniter. Sketch the necessary electric connections.

27. Explain carefully the series of operations in a two-cycle gas engine. What are the advantages and defects of this type?

28. Describe the Diesel cycle and draw an indicator card illustrating the processes.

29. Describe three methods of governing by the variable-impulse system and discuss their relative advantages.

30. What do you understand by timing the ignition, and by lead of the ignition? What is the effect of too small lead and what of too great lead? Illustrate by indicator cards.